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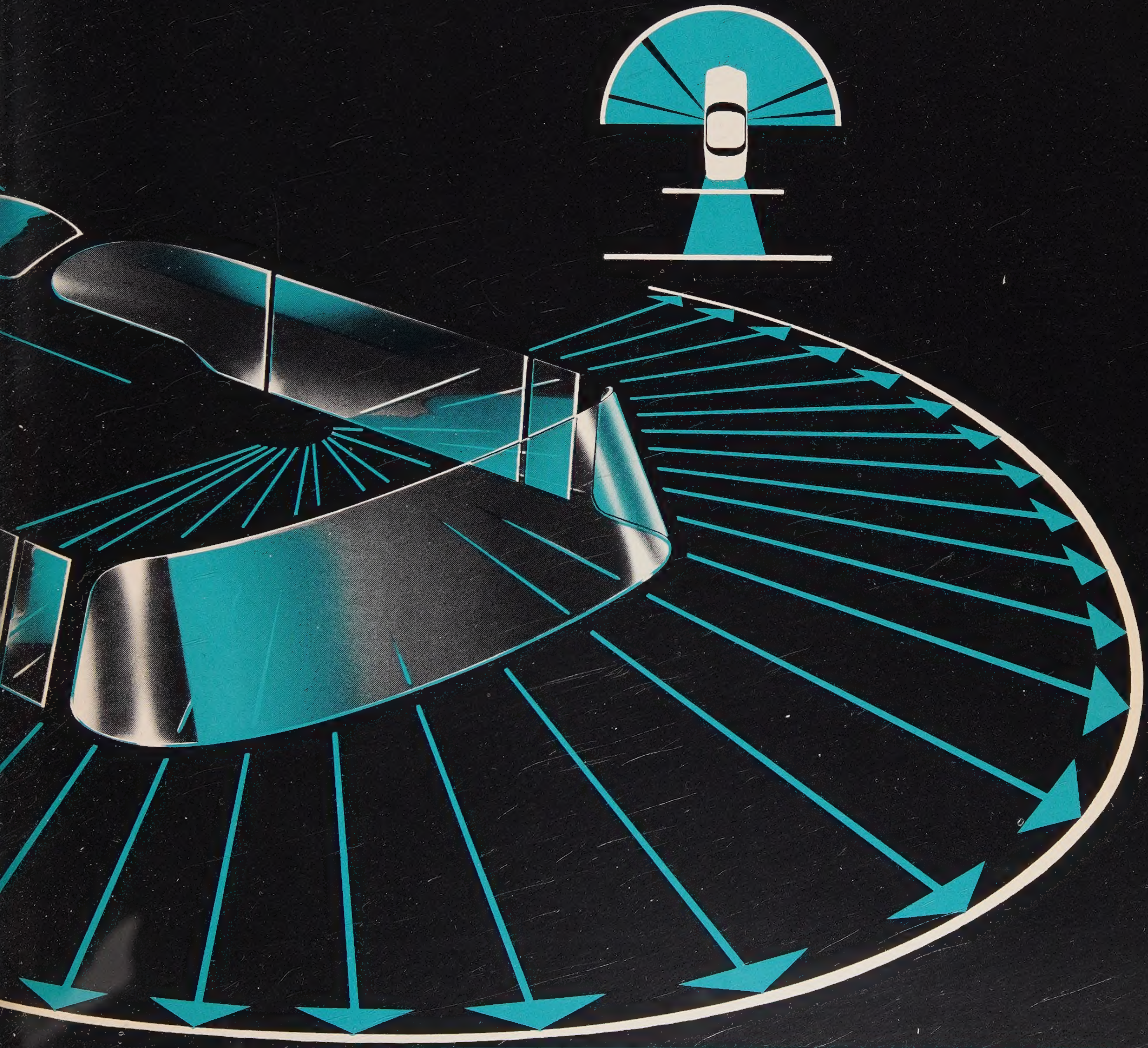
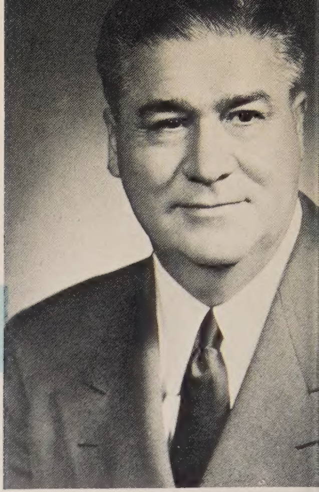


ENGINEERED VISIBILITY

On Ingenuity



When a small boy takes it into his head to make something—a boat, an airplane, a model car—he gives little thought to method. If he wants it enough, his urge to have the thing for his own pleasure knows no barriers. He just goes ahead and makes it, devising methods as he goes along.

The product of his efforts may be crude indeed—but the process is a thing of inestimable value.

For those fumbling improvisations of the child are the first stirrings of a great potential. They are the results of tension, in a budding personality, between the will to do and the want of skill—the desire to have and the lack of means.

This is the dynamics of ingenuity.

If the small craftsman perseveres, he soon finds that his most useful tool is his untrammeled imagination. He discovers, too, that the more he tries, the more facile become the hands and the mind with which he works. He has stumbled upon the formula that every industrial management in the world today is striving to use.

That is the great significance of a boy at work.

It is vitally important that young craftsmen should persevere—for they have in them the makings of a greater tomorrow—for industry, for America, for mankind.

The key to the preservation of this priceless asset of the nation is the realization that ingenuity is a growth that has deep roots—roots that take hold in childhood. It must be cultivated in childhood and nurtured through adolescence, if it is to mature in the man.

There is no short cut to ingenuity. It has either grown through the formative years—or a man has let it wither and perhaps even die. Look through the courses offered by our universities and our great technical colleges. You will not find listed a formal course in Ingenuity—but professors have the opportunity to encourage its growth in many courses.

Whether encouraged or not, ingenuity is one of our critical national resources. It is the ingredient that transforms engineering into a creative function—that advances technology into unexplored areas—that opens new doors for the industrial designer—that puts new magic into mass-production processes.

We could not build the complex, advanced-style automobile bodies of today with the tools of yesterday. It took not only highly trained men, but men of restless imagination and endless resourcefulness to devise the high-speed, precision tooling we are now using.

Basically, ingenuity is an attitude of mind, an angle of approach to new problems. It can be cultivated in young minds, free of standardized ideas. Engineering students can cultivate it well under the guidance and stimulation of professors who view students as more important than courses.

This has been demonstrated in youth programs sponsored by industry—such as the Fisher Body Craftsman's Guild model car competition—in which boys design some of their own tools out of odds and ends, make scholarship-winning model cars out of old materials and use common pins to make realistic wire wheels.

Every normal boy has in him the seeds of ingenuity. This great national resource should be cultivated assiduously.

J E Goodman

J. E. Goodman,
Vice President
of General Motors,
General Manager of
Fisher Body Division

THE COVER

The panoramic visibility designed into today's advanced automobiles is dramatically portrayed in this John Tabb design, another in a series on new developments in transportation. This critical contribution to beauty of design, motoring enjoyment, and sweeping freedom of vision for drivers represents important technical advances in several fields. Years of research and experiment were required to develop a grade of glass and manufacturing facilities suitable for the mass production to close tolerances of curved plate safety glass. Stress engineers, as well as design and process engineers, advanced the work of the stylists by developing structures that would permit slenderizing body pillars for fuller vision without sacrificing rugged strength. Finally, production engineers made their indispensable contribution in the development of assembly equipment to permit precision manufacturing of the new bodies with so much better vision all the way around.

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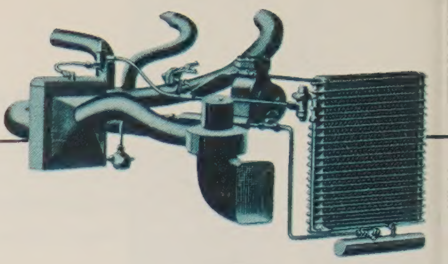
Development of an Automobile Air Conditioning System for Underhood Installation



The design and development of a mechanical refrigeration system specifically suited to meet the severe operating conditions required for automobile air conditioning application involve the solution of many complex problems. Weight, space, and ease of installation are just a few of the many factors influencing the overall design criteria of the system. In 1953, Harrison Radiator Division engineers, working in conjunction with Pontiac Motor Division engineers, began the design and production development of an air conditioning system to be introduced as an accessory item on the 8-cylinder 1954 Pontiac. The ultimate goal of producing an efficient, practical, and relatively low cost automobile air conditioning system depended upon successfully satisfying the major specification—locate the entire system beneath the underhood area.

The automobile air conditioner which provides comfort cooling during the hot weather months employs the same basic mechanical refrigeration system component parts that are found in commercial installations—but there the similarity ends. The severe operating conditions imposed upon the automobile refrigeration system while providing cooled air at proper temperature and humidity necessitate completely different design criteria than those normally encountered in commercial systems.

Three factors which influence the overall design of the automobile refrigeration system are the cooling load, space, and weight requirements. Their effect on the system is such that each individual component part must be specifically designed in order to operate at the most efficient level.

There are many variables involved in determining the cooling load requirement, such as the number of passengers, car speed, wind direction, sensible heat gain from the engine through the firewall, and solar radiation. Each variable must be given careful consideration so that the overall system will be properly sized and of adequate capacity. One of the requirements of an automobile air conditioning system is that it be able to quickly lower the temperature of the passenger compartment after the car has been standing for a long period of time in the hot sun. This means, of course, that the refrigeration system must have enough capacity to meet this requirement, in addition to providing the cooling required for low and high car speed operation.

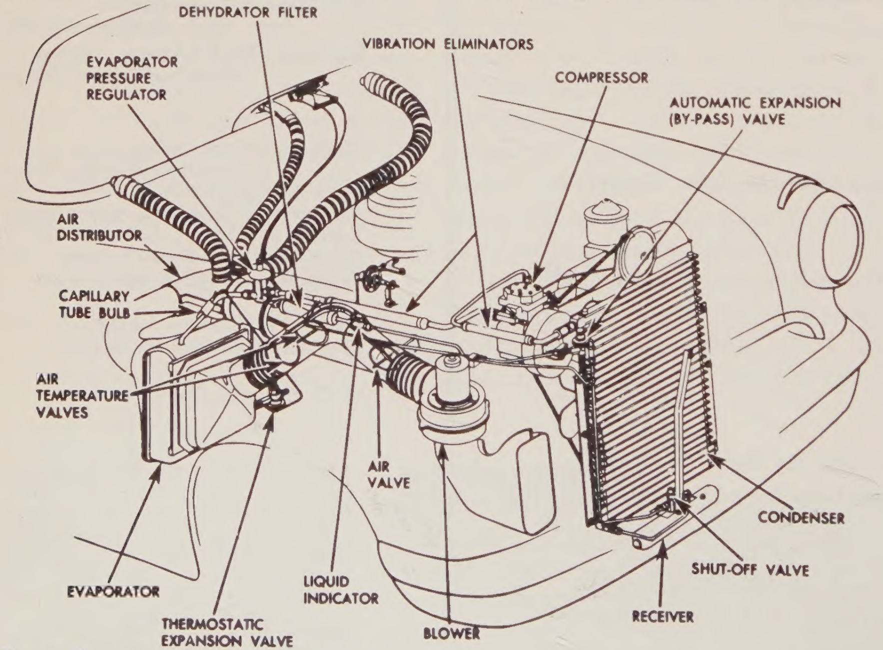


Fig. 1—The front-end type of automobile air conditioning system installation has all component parts of the system located in the underhood area. The space allotted to such an installation places severe restrictions on the overall design.

Inasmuch as the automobile air conditioner is an accessory item, the automobile designer left little room for such an installation. Space availability, therefore, is at a premium and requires a refrigeration system design which is compact and, at the same time, easily accessible for quick inspection and servicing.

Automobiles are designed to be competitive in performance—performance being a function of the engine displacement per ton-mile traveled. The automobile refrigeration system, therefore, must be as light weight as possible but care must be exercised to design the component parts strong enough to be unaffected by vibration and impact. It also is essential that the system's component parts be free from vibration which could be transmitted to the vehicle and hence to the passengers.

There is one more important consideration which must be given to the design of the automobile air conditioning system. This is a consideration involving the resultant temperature from the system. This temperature—which must be able to adjust the comfort for any number of passengers, independently, and to a satisfactory degree—requires the adjustment of an unlimited number of dry-bulb temperatures and combinations of conditioned air velocity.

The above factors and considerations served to establish the basic design criteria for Harrison Radiator Division en-

By J. RALPH HOLMES
Harrison
Radiator
Division

How system components

were re-designed for

underhood installation

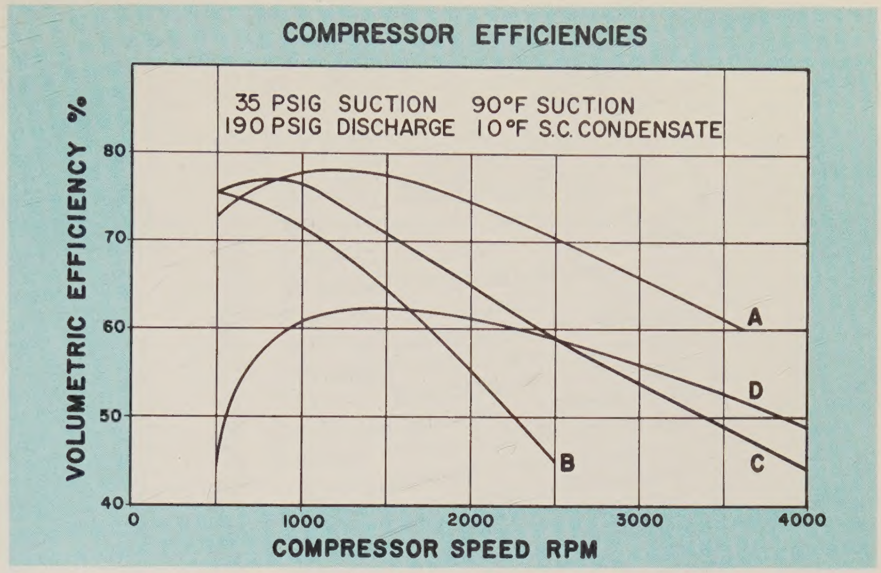


Fig. 2—This graph shows volumetric efficiency curves of four reciprocating compressors tested during the early stages of the compressor development program. In order to meet the capacity requirement of the compressor, the decrease in volumetric efficiency with increase in compressor speed had to be corrected.

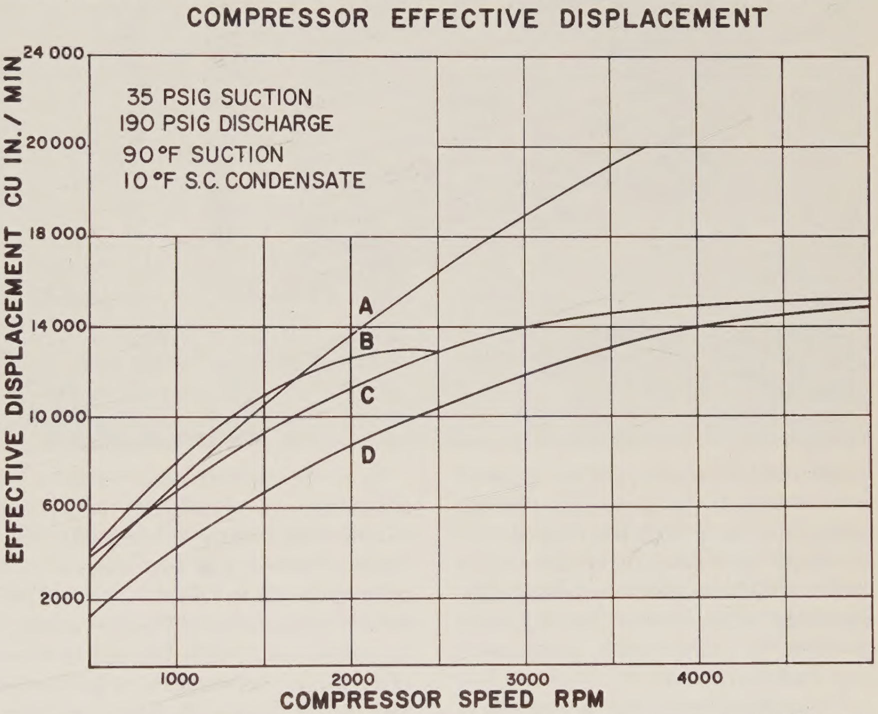


Fig. 3—Early calculations pointed out that the compressor must develop a minimum of 10,000 actual cubic inches of displaced Freon 12 per minute at a compressor speed corresponding to a car speed of 30 mph. This graph resulted from performance tests carried out on compressors having various effective displacements to meet this specification. The main problem was to develop the compressor to a point where it could be driven fast enough to displace the 10,000 cu in. It also was necessary to develop the compressor to insure that the decrease in volumetric efficiency with increase in compressor speed did not reduce the total capacity when the compressor was driven at a car speed above 30 mph.

gineers who, in 1953, began the development of an air conditioning system to be introduced as an accessory item on the 1954 model of 8-cylinder Pontiac automobiles. Harrison Radiator engineers, working closely with Pontiac Motor engineers, based the overall design on one specification—confine all component parts of the air conditioning system to the underhood area.

This type of installation differed from the then current practice of utilizing trunk compartment space to hold the evaporator of the refrigeration system. The projected design would result in a savings of material and weight by cancelling the necessity of running the refrigerant liquid and suction lines from the compressor and condenser which were normally located in the underhood area. The stringent space limitations imposed upon this front-end type of refrigeration system installation introduced many problems which required an extensive developmental and testing program before the final design of the air conditioning system shown in Fig. 1 was realized.

Compressor Development

The outline of the compressor development program was based upon testing various commercially available compressors with a view to establishing information which would form the basis for a specific compressor design suitable for automobile application. Information upon which the overall design criteria was based was gathered from the testing of various 2-cylinder and 4-cylinder vertical and V-type reciprocating compressors having piston displacements of 6.63 cu in., 8.12 cu in., 8.63 cu in., 9.2 cu in., 11.4 cu in., and 12.7 cu in.

From the studies and tests made on the various compressors it was decided to take a 2-cylinder, vertical, single-stage, in-line, reciprocating compressor and re-design it for automobile application. The re-design of this compressor included entirely new valve structures, lubrication system, and the use of ball bearings on both ends of the crankshaft. The crankcase also was re-designed to be adaptable for either vertical or horizontal mounting and clockwise or counterclockwise rotation. By doing this, the same compressor-body casting could be used for either of the mounting or rotation conditions.

The prime compressor-design consideration, in addition to minimum size and

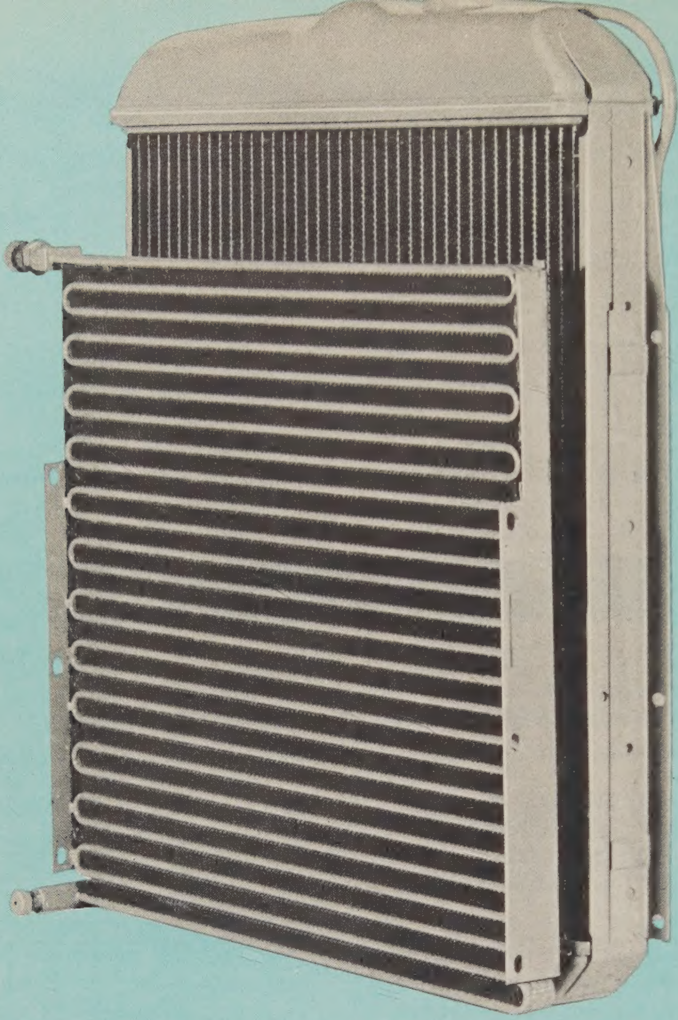


Fig. 4—The final condenser and radiator assembly design uses a shallow depth, large frontal area, steel-tube condenser which is copper brazed at 2,000° F to give an efficient thermal and structural joint.

weight considerations, was the capacity requirement. It was imperative that the compressor have sufficient capacity to distribute the refrigerant, which was to be Freon 12, throughout the system while operating under widely varying head pressures, suction pressures, speed ratios, and load conditions.

The capacity of a compressor is a function of two factors—its speed and *actual effective displacement*, which is expressed as a specific quantity of refrigerant circulated per unit of time. The actual effective displacement, in turn, depends on the compressor's bore, stroke, number of cylinders, speed, and volumetric efficiency. The first step taken to meet the capacity requirement was to rebore the 2-cylinder compressor to increase the piston displacement from 8.12 cu in. to 8.63 cu in. This was the largest bore that could be used with the particular crankshaft design of the compressor.

The reboring operation reduced the remaining variables affecting the capacity to two—speed and volumetric efficiency. Results from various tests performed on compressors showed that volumetric efficiency decreased rapidly after reaching a maximum value, with increase in speed (Fig. 2). If the capacity requirements were to be met on the particular compressor selected it was necessary that the decrease in volumetric efficiency be effectively corrected at this condition of increased compressor speed.

A definite value for capacity, in terms of actual effective displacement and compressor speed, had been established at the start of the compressor development program. Calculations indicated that the compressor must develop an actual effective displacement of 10,000 cu in. of Freon 12 per minute at a car speed of 30 mph. (The factor, actual effective displacement, is arrived at by multiplying the swept volume of the compressor's pistons, in cubic inches, by the volumetric efficiency at the speed it is intended to be driven.) The ability of the compressor to meet this displacement requirement resulted from performance tests which were carried out on compressors having various effective displacements (Fig. 3). The 2-cylinder compressor which was selected was developed to a point where it could be driven fast enough to displace 10,000 actual cubic inches of Freon 12 per minute at a car speed of 30 mph and maintain this capacity or a greater capacity at all higher car speeds.

The volumetric efficiency of the compressor at a car speed of 30 mph was high enough so that the rotational speed of the compressor was at a reasonable value. This served to establish a ratio between engine speed and compressor speed which carries on up to the maximum road speed of the car, established as a general factor of 105 mph. The ratio was such that if the air conditioning system was to be operated while the car was traveling at 105 mph, the compressor would have to run at a speed of approximately 5,000 rpm. This extremely high rotational speed necessitated further developmental work in order to insure that the compressor would have the required durability.

Compressor Drive and Lubrication

The compressor, which is mounted on the top-front, right-hand side of the engine cylinder head by a steel bracket, receives its source of motive power from the crankshaft of the car's engine by means of a V-belt drive. A special pulley is attached to the harmonic balancer of the engine crankshaft and the V-belt runs from this pulley upward to the compressor and then across to a compressor-belt idler pulley which is held in place by means of a steel bracket braced to the engine with struts that extend diagonally down to the engine front-end plate. The V-belt is tensioned by an upward and outward rocking of the idler pulley.

Harrison Radiator engineers decided at the start of the developmental program that the air conditioning system should not constitute a drain on engine power and gasoline consumption when comfort cooling was not required. This meant that a method had to be devised for disconnecting the compressor from its source of motive power. As a result, an electrical disconnect clutch was developed and made an integral part of the compressor's drive arrangement.

The compressor clutch utilizes an electromagnetic coil which pulls the clutch driven plate into engagement with the clutch surface of the coil housing. This action serves to drive the compressor. The electromagnetic coil of the clutch receives electrical power by means of a slip ring and brush assembly positioned beneath the compressor and clutch. Electrical power is delivered only when the air conditioning system is turned on.

The lubrication system for the compressor is arranged as an integral part of the refrigeration system. The lubricating oil used is compatible with the Freon 12 and is placed in the crankcase of the compressor during assembly. The ability of the Freon 12 to be compatible with the lubricating oil allows the refrigerant to serve as a cooling medium for the compressor.

To prevent the possibility of improper lubrication during conditions of low Freon 12 flow which occurs during light load operation, a piping and valve system was arranged to insure the existence of sufficient lubricating fluid for the compressor. A by-pass valve is placed in a line connecting the discharge side of the receiver to the suction line returning from the evaporator to the compressor. When the compressor suction pressure drops below 15 psi the by-pass valve opens and permits a metered amount of refrigerant and lubricating oil mixture to return to the compressor directly from the receiver.

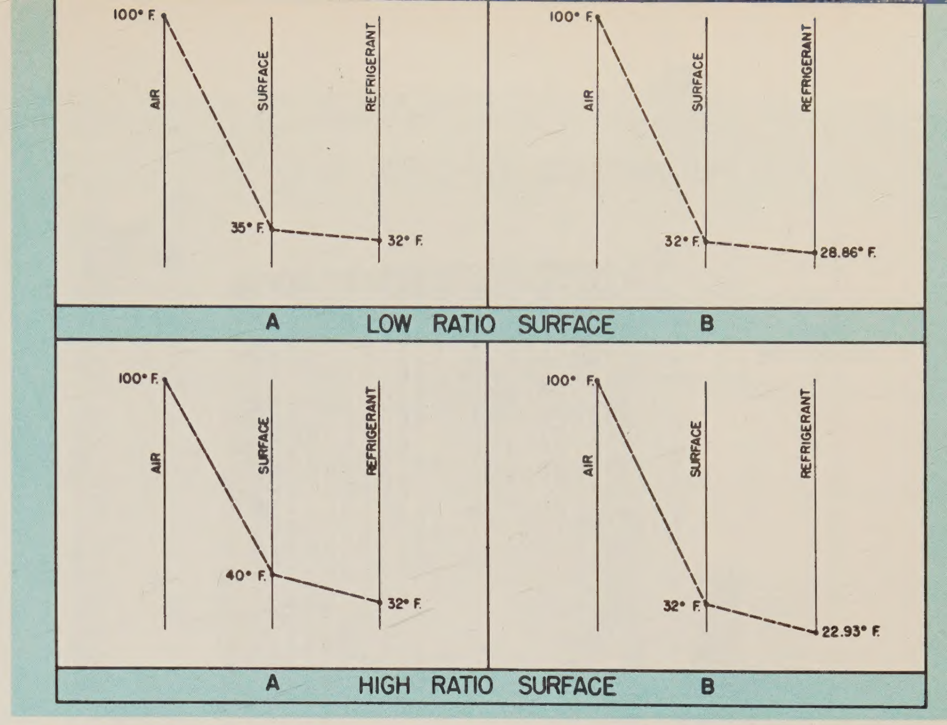


Fig. 5—The theoretical design of the evaporator was based upon the selection of one of two principles—make the air side and Freon 12 side of the evaporator *strong* (low-ratio surface) or make the air side *strong* and *weaken* the Freon 12 side of the evaporator (high-ratio surface). The low-ratio surface principle was selected and required careful control of the evaporator-surface temperature close to 32° F in order to prevent freezing of the moisture condensed out of the warm air while passing through the evaporator.

Condenser Development

The function of a condenser in a refrigeration system is to liquefy and reduce the temperature of the hot refrigerant discharge gas received from the compressor by means of a cooling medium. The type of condenser which lends itself best to automobile application, and the one used by Harrison Radiator engineers, is of the air-cooled, dry type. The logical location for the condenser, in order to take full advantage of the air supply for cooling the refrigerant gas, is in the space between the engine radiator and grille.

There were many factors which were considered during the condenser design stage. The function of the condenser made it mandatory that it effectively satisfies all of the fundamentals of good heat transfer efficiency. The operating conditions to which the condenser would be subjected required a construction capable of withstanding high pressures which were necessary if the Freon 12 was to be condensed at the outside air temperatures normally encountered. The location of the condenser required that it be compact and light weight. Another factor which had to be considered in regard to the location was that the condenser not cause serious air obstruction to the radiator and thereby decrease the efficiency of the engine cooling system.

The various factors affecting the condenser design necessitated considerable development and testing work before a final condenser design was achieved. Fig. 4 shows a view of the condenser and radiator combination now being used.

The condenser design which was finally developed utilizes a shallow depth and a large frontal area. Four parallel serpentine tubes, ¼ in. in diameter, wind back and forth with the indirect, or air-side surface, between each convolution of tubing. The entire condenser assembly is brazed with copper at approximately 2,000° F to effect an efficient thermal and structural joint. This provides a very homogeneous and high-pressure unit requiring minimum hand labor for its final assembly. The tubes and air-side surface of the condenser are of steel.

Refrigerant Auxiliaries

The refrigerant, in flowing from the condenser to the evaporator, passes through four major auxiliaries necessary for the proper operation of a refrigeration system. The liquid refrigerant flows from the condenser into a receiver which serves the purpose of storing refrigerant if the compressor is pumping more refrigerant than the evaporator can consume. The receiver also assures that the evaporator will have a sufficient supply of refrigerant to provide cooled air immediately after the air conditioning system is turned on. The design of the receiver had to include provisions to insure that it would maintain a sufficient quantity of refrigerant to take care of changes in ambient temperature.

From the receiver the refrigerant flows through a sight glass and then through a filter and dehydrator assembly before reaching the thermostatic expansion valve. The sight glass provides a means for inspecting the quantity of refrigerant in the system. Presence of bubbles in the flow denotes that the refrigerant is reaching the minimum usable level. The filter and dehydrator remove any solid matter or water vapor trapped in the system. The thermostatic expansion valve con-

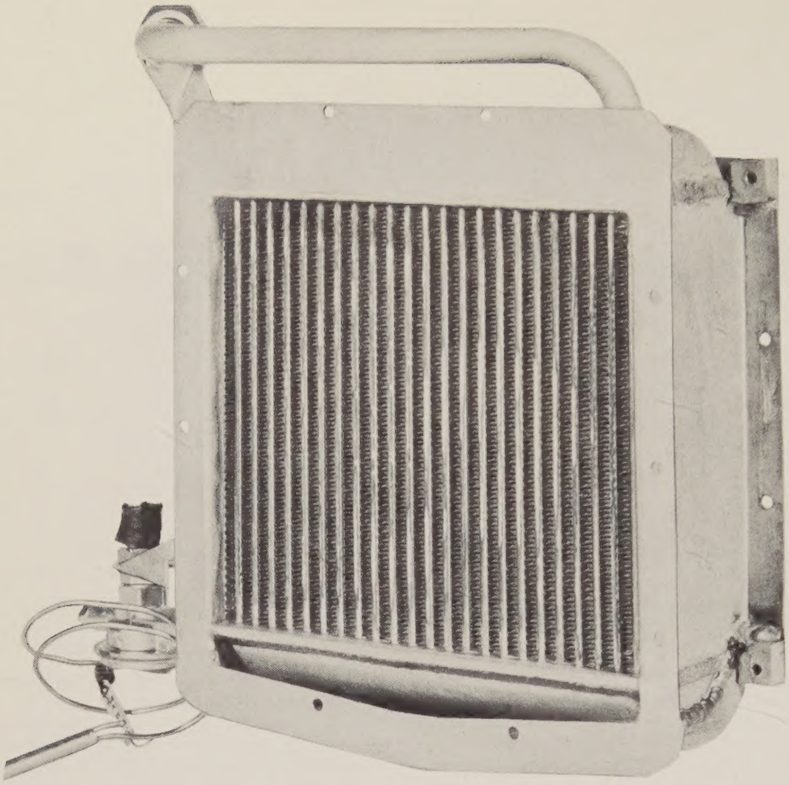


Fig. 6—To meet the weight requirement, an aluminum evaporator was developed. Normally, aluminum is not considered a durable metal when used in a Freon refrigerant system, due to the presence of alcohol injected into the system to prevent freezing of the thermostatic expansion valve orifice. Through the development of a special control system which prevents such freezing, the use of alcohol is not needed.

trols the flow of refrigerant into the evaporator and greatly reduces the pressure and, thereby, the temperature of the refrigerant.

Evaporator Development

The function of an evaporator in a refrigeration system is to remove heat from the medium being cooled (air) and allow effective boiling of the refrigerant with a minimum pressure drop through the evaporating unit's coils. Of the two general types of evaporators in use, dry and flooded, Harrison Radiator engineers elected to use the flooded type. With this type of evaporator all of the refrigerant is not evaporated. A liquid-vapor refrigerant mixture leaves the evaporator and flows into a surge drum. The vapors are then drawn into the compressor suction line and the liquid recirculated within the evaporator.

The overall design of the evaporator involved the consideration of many factors. The evaporator had to be compact and light weight. Because it had to be compact, the heat-transfer surface had to be highly efficient on both the liquid and air side. A highly efficient heat-transfer surface on the air side of the evaporator would permit an overall reduction in size and allow required air flow. The evaporator had to be capable of easily removing the water condensed from the air, which in the case of 100 per cent outside air would be considerable during humid weather operation. The evaporator had to be designed to prevent plugging up under dust-condition operation. Lastly, the system controls had to be designed so as not to freeze the condensate in the evaporator while the compressor was delivering its maximum capacity.

The theoretical design of the evaporator was governed by two principles—one of which had to be decided upon before the overall design could begin. The first principle called for making the air side of the evaporator *strong* and the Freon 12 side *strong* (low ratio). The second principle was to make the air side *strong* and *weaken* the Freon 12 side (high ratio).

The low-ratio design principle necessitates close control of the Freon 12 side of the evaporator when the compressor speed is such that it removes vapors faster than the thermostatic expansion valve delivers refrigerant. Under this condition, the pressure in the system drops and causes a reduction in the boiling point of the Freon 12. This, in turn, causes a lowering of the evaporator surface temperature below the freezing point of the moisture being condensed out of the warm air and results in evaporator "freeze-up."

The high-ratio design principle permits a reasonable performance at low compressor speeds when the Freon 12 side of the evaporator is weak and, at the same time, permits the capacity of the evaporator to build up with an increase in compressor speed. This is due to the fact that the temperature of the evaporator metal on the weak Freon 12 side does not drop to the boiling point of the Freon 12 when the air side is relatively stronger than the Freon side.

The high-ratio design principle, after careful consideration, was discarded in favor of the low-ratio principle. Harrison Radiator engineers felt that the high-ratio design principle would require a larger size compressor than the low-ratio principle, in that it would have to deliver the same tonnage at lower density.

Fig. 5 shows theoretical temperatures of air, evaporator metal surface, and refrigerant determined during the evaporator development stage and used for final design purposes.

To control the surface temperature of the evaporator core close to 32° F, which is the approximate temperature of the Freon 12 after passing through the thermostatic expansion valve, a back-pressure regulator valve is used. This valve controls the flow of Freon 12 and throttles the refrigerant discharge from the evaporator to a practically constant pressure of 32 psi. If the pressure dropped below this value the temperature of the evaporator core would fall below the freezing point of water. The moisture condensed out of the warm air passing through the core would form ice in the air passages and, therefore, would restrict the air flow and reduce the capacity of the evaporator.

The weight factor of the evaporator received serious consideration and, as a result of a concentrated metallurgical study, an aluminum evaporator was developed (Fig. 6). In the commercial refrigeration field, aluminum is not considered a durable metal to be used with a Freon refrigerant system. This is due to the presence of alcohol which is injected into the system to prevent freezing of the thermostatic expansion valve orifice. The aluminum evaporator was successfully used in the Harrison Radiator design because of the development of a control system which prevents freezing of the orifice, thereby eliminating the need for alcohol.

Radiator Correction

After all components of the refrigeration system had been developed, wind tunnel tests were made to determine what effect the system had on the engine-coolant system.

Fig. 7 shows a graph resulting from data which were established during this study. The graph shows car speed, in miles per hour, plotted against an *air-to-boil index*, which is the term used to designate the temperature at which the water in the radiator boils at any speed of the car. In theory, it is desirable to have the air temperature, at which the engine-coolant boils, the same at all speeds of the car. In actual practice, however, this is not possible and, as a result, a satisfactory temperature must be obtained at low car speeds and the increased temperature at the higher speed accepted. It is for this reason that the low car speeds were critical and received more attention during the study.

From Fig. 7 it can be seen that a standard car equipped with a standard radiator core and fan and without an air conditioning system installed had an air-to-boil index of 110° F at a car speed of 20 mph. When the air conditioning system was installed, the air-to-boil index decreased to 95° F at 20 mph. This temperature was a critical value and a correction had to be made. The correction made was in the form of adding more copper to the radiator core. This served to raise the air-to-boil index to 103° F at 20 mph. This new value, however, was still below the original cooling index of the standard car with a standard radiator core and fan. In order to take full advantage of the additional amount of copper which was added to the radiator core, the fan was altered to provide an increase in air flow through the radiator. This alteration served to raise the air-to-boil index to a satisfactory 117° F at a car speed of 20 mph.

Tests also were conducted to determine the effect of the air conditioning system on engine radiator boiling during engine idling speed operation. The results of the investigations are tabulated at the bottom of Fig. 7.

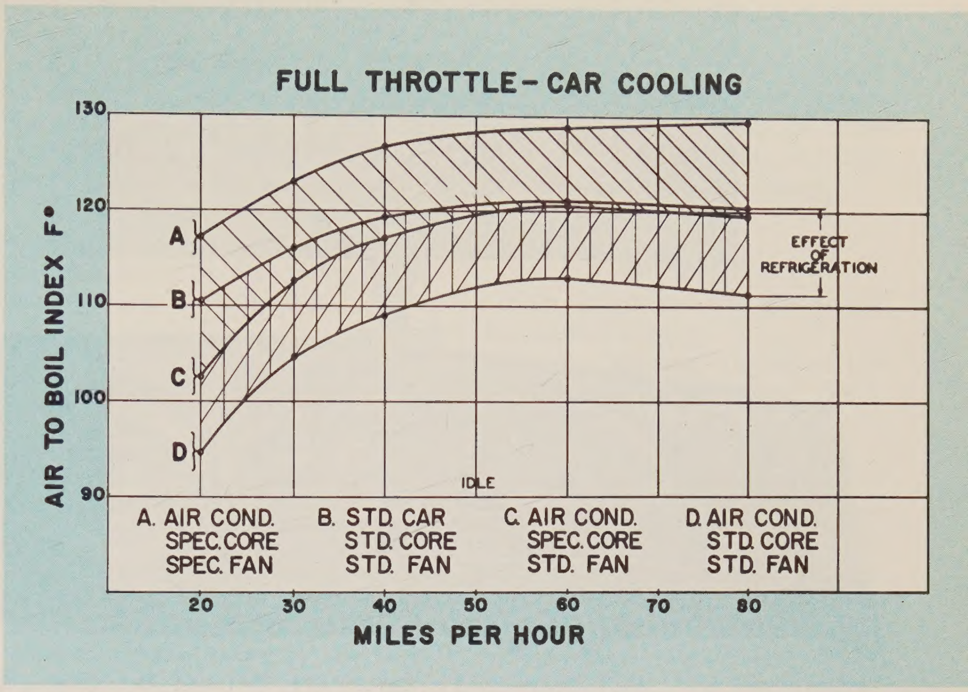


Fig. 7—Special tests were conducted after all component parts of the refrigeration system were developed to see what effect the system had on the efficiency of the engine-coolant system. When a standard car with a standard radiator core and fan and with no air conditioning system installed was tested, the radiator boiled at a temperature of 110° F at a car speed of 20 mph. When an air conditioning system was installed the radiator then boiled at 95° F at a car speed of 20 mph. To correct the lowering of the temperature at which the radiator boiled, a special radiator core was first used and then a special six-blade, high-pitch fan was added which aided materially in raising the radiator boiling point to 117° F at a car speed of 20 mph, which compared favorably with the original cooling index of the standard car without air conditioning. Tests also were run to determine what effect the air conditioning system had on radiator boiling when the system was in operation under idling conditions. The first test showed that the radiator boiled at 230° F in 33 minutes when the engine idled at 400 rpm in 100° F ambient air. To correct this condition a fast-idle mechanism was used which increased the engine idle speed to 750 rpm and improved radiator performance by having the radiator temperature stabilized in 60 minutes at 215.3° F when the car was idled in 118° F air while using a special radiator core and special fan.

When a standard car equipped with a standard radiator core and fan and no air conditioning system was run at an engine idling speed of 400 rpm in 100° F air, radiator boiling occurred in 33 minutes at a temperature of 230° F. It was realized that if the air conditioning system would be used while the car was idling there would be sufficient obstruction to the air flow into the radiator to seriously affect the operation of the engine-coolant system. In view of this, a fast-idle mechanism was employed which aided materially in providing adequate engine cooling and in affording increased efficiency to the air conditioning system when operated during idling speed.

The *fast-idle mechanism* is an electrically controlled vacuum-powered diaphragm unit that sets the carburetor throttle to give an engine idling speed of 750 rpm when the transmission is in neutral. When the unit is energized electrically, a solenoid operates a vacuum valve which causes a vacuum diaphragm to open the engine throttle to give the idle speed of 750 rpm.

When a car equipped with air conditioning, special radiator core, standard fan, and the fast-idle mechanism was tested under idling conditions in 100° F air, radiator boiling occurred in 1 hr at a temperature of 225° F. When this same car was again tested, but with the fan of increased capacity, at a temperature of 118° F air, radiator stabilization occurred at the end of 1 hr at a temperature of 215° F. This last factor is highly important when it is considered that there is no way of knowing what the outside air temperature, sun load, and wind direction will be when air conditioned cars are idled. By having a high stabilization point, the air conditioning system operates more efficiently when operated under engine idling conditions.

Air Distribution System and Controls

The overall design of the air distribution system was based upon satisfying three conditions. The duct work for the intake and supply air had to be located in such a manner as to use most advan-

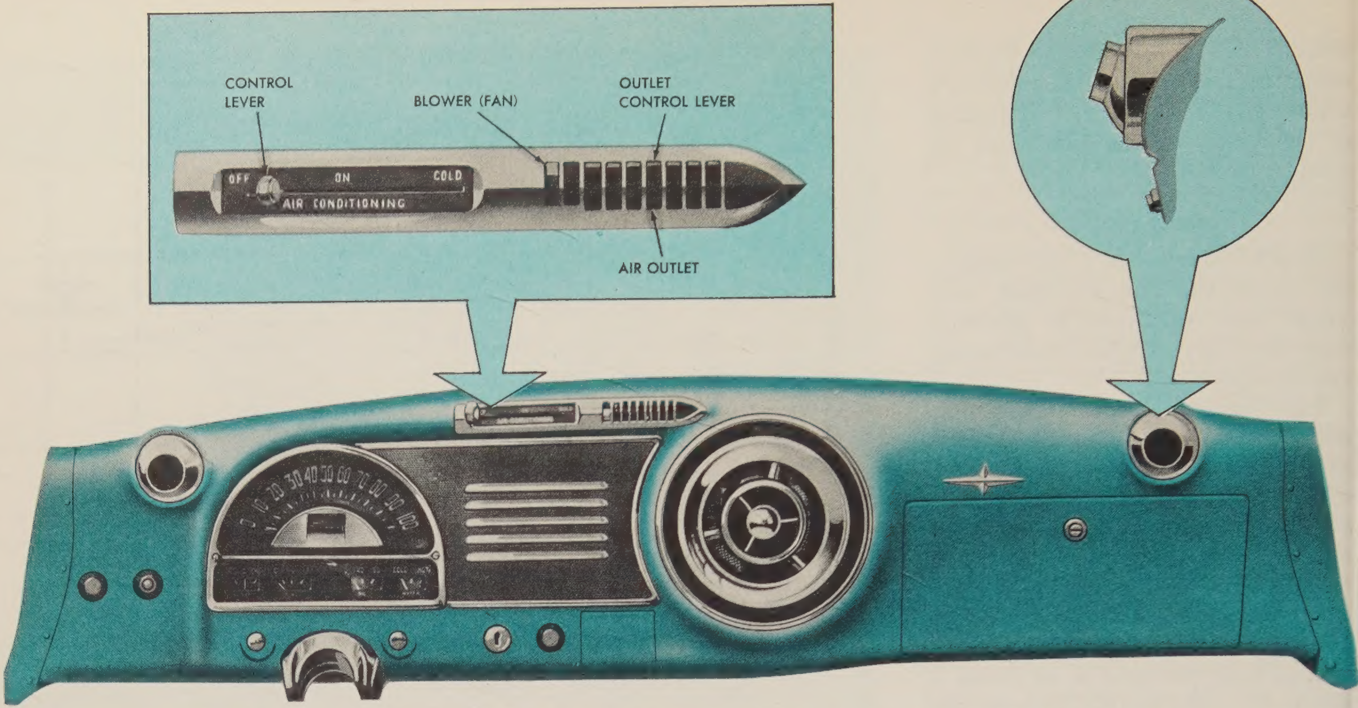


Fig. 8—Overall view of the discharge outlets and control panel as developed for the initial installation on the 1954 8-cylinder Pontiac. The driver controls the amount of air passed through the evaporator from the control panel lever, shown in the upper left inset, which operates dampers in the duct work. Movement of the lever to the right causes a greater quantity of outside air to pass through the evaporator and results in providing maximum cooling for the particular speed at which the car is traveling. The blower also is controlled from the control panel and may be used for forced intake-air flow during engine idling or slow speed operation. The two outer discharge nozzles are mounted in ball sockets and the third discharge nozzle, which has a valve deflector plate with a shut-off valve, is located in the right side of the control panel. A similar arrangement is used for 1955 models.

tageously the restrictive space limitations of the underhood area. The method for controlling the supply of conditioned air had to be accomplished with a control mechanism simple and easy to operate. Lastly, the conditioned air discharge outlets had to be strategically located to provide the passengers with conditioned air in the most effective manner and without draft. Fig. 8 shows an overall view of the final design of the duct work, control panel, and air discharge outlets.

The design of the overall air conditioning system is based upon providing the passengers with the option of as much new air as possible. This provision results in providing comfortable air conditions inside the passenger compartment while under full load. The 1955 air conditioning system design, as used on the Pontiac and Oldsmobile cars, has provisions for supplying some recirculated air, in addition to the above feature.

The duct work for the outside air supply is designed so that the driver has the option of utilizing all or only a portion of the available outside air for cooling purposes. This is made possible through the use of dampers which are operated by the driver from the system's control panel. In the event that recirculated air is desired, the control panel is designed in such a manner that the control lever, when moved to a certain position, closes the outside air supply and opens an air passage from the passenger compartment leading to the blower and then to the evaporator.

Outside air enters through a removable and cleanable filter behind the radiator and passes upward through the blower. At the blower outlet the duct work splits into two sections—one duct running to the evaporator and the other running to the air-distributor chamber. A third duct is used to connect the evaporator outlet to the air-distributor chamber. The flow of air to the passenger compartment is obtained by the use of three flexible hoses connecting the air-distributor chamber to the instrument panel outlets.

The quantity of outside air passed through the evaporator determines the amount of cooling effect given the air supplied to the passenger compartment. The driver controls this air flow through the evaporator from the air conditioning system control panel, which utilizes a horizontally sliding lever to manipulate two bowden cables which operate as a function of each specific setting of the lever. When the control lever is moved from the OFF to the ON position, the first bowden cable is put into operation and opens an electrical master switch which energizes the magnetic clutch and puts the compressor into service. This bowden cable also opens the main outside-air valve which permits outside air to enter the air distribution system. When the control lever is moved from the ON position to the COLD position the second bowden cable is brought into operation and allows the mixing of air between the evaporator and the by-pass air duct by controlling the damper positions in the evaporator-air duct and the by-pass air duct according to the position of the lever. Continued movement of the control lever to the right causes a greater quantity of outside air to pass directly through the evaporator. When all of the incoming air is passing through the evaporator, maximum cooling results for the particular speed at which the car is traveling. The 1955 design of the air distribution system has provisions made for recirculated air by adjustment of control levers which close the outside air passage 90 per cent and open an air passage from the passenger compartment to the blower, thereby resulting in recirculated air to be passed to the evaporator.

The split-duct arrangement affords a wide range of temperature control and permits the driver to change the temperature of the air coming into the passenger compartment from approximately 3° F above ambient conditions down to the full capacity of the refrigeration system which, at 100° F dry bulb and 40 per cent relative humidity, is approximately 50° F. The 3° F temperature above ambient conditions results from the temperature pickup from the air entrance to the air-discharge outlets in the passenger compartment when the air is by-passing the evaporator.

If the air conditioning system is operated during slow car speed or idling conditions the motor-driven, low or high speed, blower may be used for forced intake-air flow. The blower, operated from the control panel, is designed to give a large volume of air flow and creates sufficient static pressure to force the incoming outside air through the many restrictions in the air distribution system. The electrical circuit for the blower motor becomes energized when the air conditioning system control lever is moved from the OFF to the ON position. Through the use of the blower and the fast-idle mechanism, the refrigeration system is able to operate at greater capacity when cooled air is desired while the car is stationary.

The two discharge nozzles located at the right and left end of the instrument panel are mounted in ball sockets and are completely adjustable so that air distribution can be directed straight up and over the heads of the passengers, over the outside shoulder, along the inside roof line, or to impinge upon the side door glass should the sun be shining through that particular window. The middle nozzle, located in the right side of the control panel, contains a valve deflector plate fitted with a shut-off valve. This discharge outlet supplies a stream of cool air to the center portion of the passenger compartment and prevents the flow of air from the outer nozzles from turning back and impinging upon the backs and necks of the passengers after reaching the back seat. The air is exhausted from the car body around the windows and doors and through other exits in the car body. The air conditioning system maintains sufficient pressure inside the passenger compartment to insure that exfiltration of air is maintained and that infiltration of hot and humid air does not take place which may produce jets of unconditioned air impinging directly upon the occupants.

Wind tunnel and field tests have shown that the conditioned air distribution from the front seat to the back seat is within not more than 2° F differential.

Car Modification

To accommodate the air conditioning system, the standard 8-cylinder Pontiac required various modifications. The standard radiator core was replaced with a tube-and-center type for high-pressure engine cooling. The standard engine fan was changed from a four-blade to a six-blade, high-pitch fan. The engine crankshaft harmonic balancer was modified to provide an extra groove for the compressor V-belt drive. A heavy duty generator and voltage regulator is required along with a 19-plate battery. Heavier front springs and 7.60 by 15 tires are required for increased load-carrying capacity. Rear-axle gear ratios were changed to retain desired car performance and special instrument panel piercing was required to facilitate the placement of the air conditioning control panel.

Summary

The basic components of the mechanical refrigeration system have been successfully re-designed and adapted to an automobile air conditioning system. A compressor of minimum weight delivers the required capacity at widely varying conditions of pressure, speed, and load and with suitable control of volumetric efficiency at the higher compressor speeds. When comfort cooling is not wanted by the car passengers, the compressor is disconnected from the engine by an electrical clutch. The condenser is compact, light in weight, offers minimum obstruction to the car radiator, and has good heat-transfer efficiency. Compactness and light weight also are achieved in the evaporator which is constructed of aluminum and is based on the low-ratio design principle permitting a smaller compressor size. The easily adjusted air outlets permit the car occupants to receive cooled, dehumidified, and draft-free air at any desirable temperature through the utilization of a split-duct arrangement which is effectively controlled from a simply operated control panel.

The air conditioned car has caught the public's fancy and developmental work will continue in search of further weight reductions, lower costs, and other refinements as car models change.

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Testing the Structure of an Automobile Body

An important step in the design of an automobile body is the structural testing of the body and its components. In fact, it is sometimes said that the body engineer of today can complete a design with the assurance that practically no design difficulties or failures to be corrected lie ahead. This air of confidence can be attributed to the extensive test program which has followed the design from the time it was an idea to reality. In a structure as complex as an automobile body, any theoretical approach to actual determination of body loads must be made with judgement in the light of past experience. At the Fisher Body Division, definite standards are established based on satisfactory performance of previous designs. Therefore, to insure that new designs will be equal to or will surpass the standards, the body must undergo tests for torsion and bending as well as a jacking test and a shake-rig test and tests of various components. Final road testing of the complete car verifies the laboratory results. Since the adoption of this procedure, production delays or service difficulties arising from structural problems have been rare. Improved quality and comfort have resulted for the user.

LABORATORY testing of automobile body structures has gained wide recognition during the past few years. The General Motors car and truck manufacturing Divisions schedule a rigorous test program which they consider a prerequisite to production of a car. The Fisher Body Division of General Motors likewise has attached particular importance to structural tests. Production schedule delays and service difficulties resulting from structural deficiencies of the automobile body have rarely occurred since testing has become a part of every design program.

Structural tests involve body components, sub-assemblies, the completed body, and the entire car. For components and sub-assemblies, test setups and analyses of results follow basic engineering principles established for the loading of simple beams and columns. Tests of the frame alone, body, and the complete car are somewhat more involved. To obtain the information desired, engineers have adopted certain static tests, as well as representative dynamic tests, which have proved reliable in determining whether or not an automobile body has sufficient strength and stiffness to perform satisfactorily on the road.

Standards for the evaluation of a structure are based on the performance of past models which have proved satisfactory. In testing any component member or structure, the engineer must endeavor to obtain a test setup approximating the future use of that member. His standard—or comparison specimen—must be tested in an identical manner to assure a minimum of experimental error.

Actual body loads are unknown for the most part and are difficult to determine. In a structure as complex as the automobile body, it is evident that any theoretical approach to actual load determination must be tempered with judgement and past experience. On the road, the complete car is subjected to many variable conditions. In a laboratory test involving the body or its components, it is obvious that any reasonable estimate of loading which provides a fair comparison with the standard is the only sound approach.

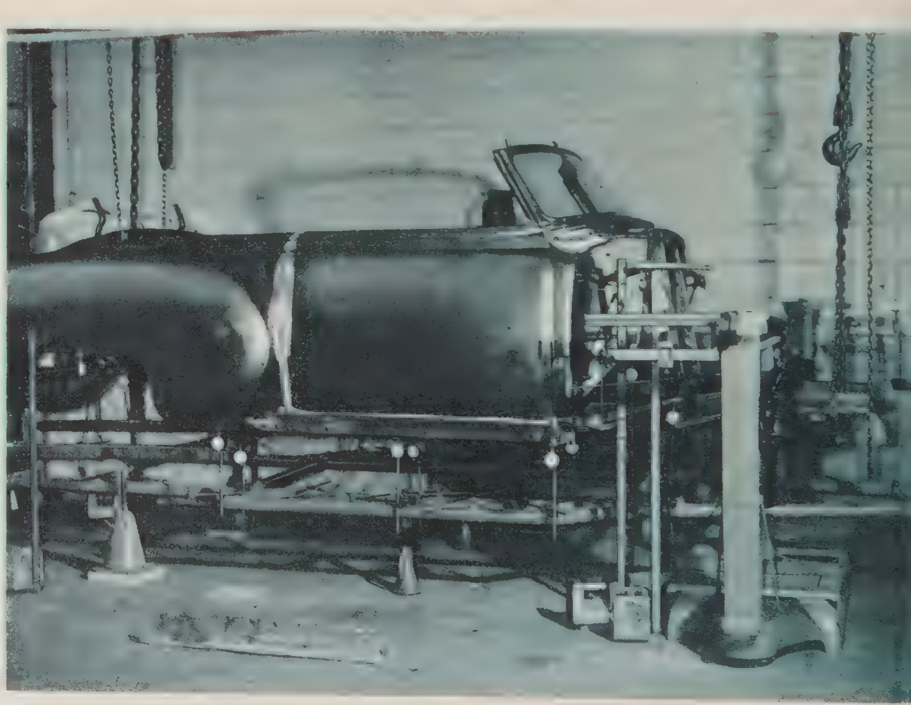

Fig. 1—This laboratory setup illustrates a body-frame combination undergoing a test for torsional rigidity. The frame is supported at the front- and rear-axle locations. A torque load may be applied to one side of a frame side rail on the centerline of the front axle by means of a hydraulic jack (partially obscured by the scale). The deflection is measured by indicators placed at specific locations along the frame side rails and rocker panels. From the observed data, engineers may calculate the torque applied and the angle of twist to determine the torsional rigidity in pound-feet per degree. These data also are used to plot the curves shown in Fig. 2. The torsion test indicates the automobile's ability to resist twisting, such as might occur when one front wheel hits a bump.

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Division

A design help: fundamental tests of the body structure

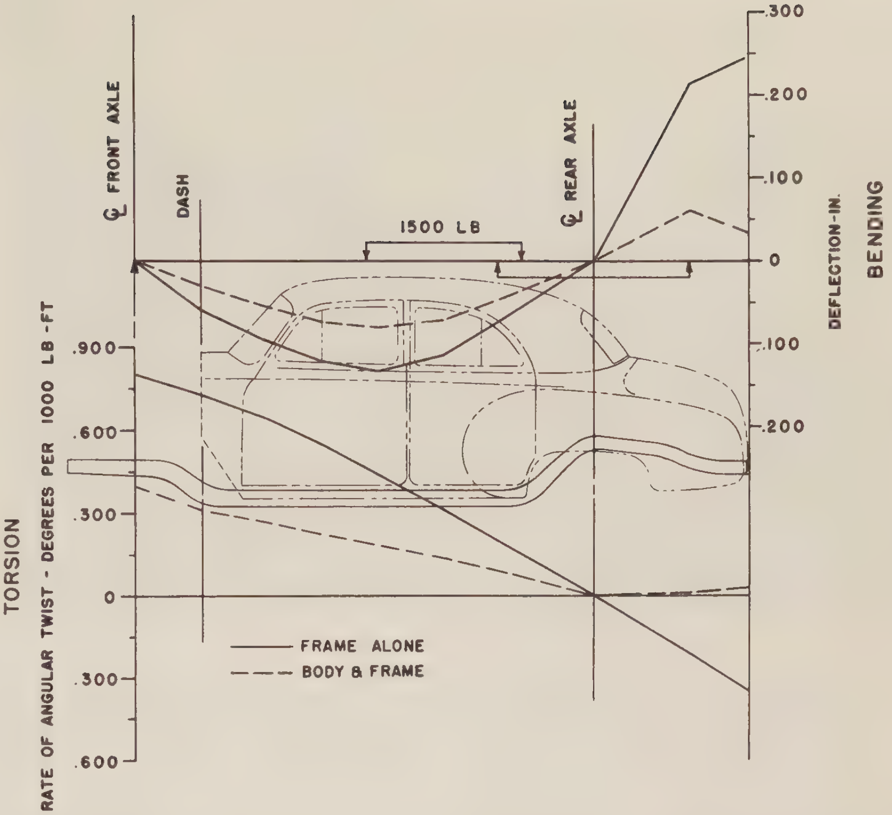

Fig. 2—The deflection and rate of angular twist of the frame alone versus the body-frame combination are compared to these curves. These curves are plotted from a load of 1,000 lb-ft using the rear axle as the origin. The results of the bending test to determine deflection also are shown by the curves assuming a static load of 1,500 lb and zero deflection in the front- and rear-axle supports. The contribution of the body to the body-frame combination is evident from these curves.

The results of static structural tests are verified through road testing of the complete car. Tests conducted during the experimental stage of body engineering are only an indication of what may occur on the road. Final proof that the original design is satisfactory for production can be obtained only by actually driving the vehicle under conditions at least as severe as those encountered by the motorist. Fisher Body and the General Motors car Divisions conduct what is known as a *durability test* to reveal any deficiencies which may have escaped the engineer during laboratory or preliminary road testing.

A typical body durability test as conducted by Fisher Body includes satisfactory completion of road tests involving severe mud and gravel conditions and 5,000 miles of rough block roads. Daily logs are maintained showing the performance of various body items, such as locks, doors, body panels, welds, and seat contours. Intermittent tests to determine rigidity of the body under beaming, torsion, and jacking conditions are conducted to prove the efficiency of the structure.

Torsion Test

The laboratory procedure for structural testing of automobile bodies follows basic fundamentals and requires little special equipment. There are several different methods used by engineers in determining the torsional rigidity of an automobile frame, body, or complete car. Basically, regardless of how the unit is twisted, the rate of twist is a function of the structure and is not influenced by variations in the testing procedure. Described herein is a method which has proved satisfactory for testing frame, body-frame combination, and complete car. In the example discussed, the method is adapted to a body-frame combination.

The contribution of the body alone to overall rigidity is usually determined through tests of the chassis frame alone and then in combination with the body. The body contribution is deduced from the rigidity of the combination. The mountings or *shims* may have an effect on the rigidity of the combination and, therefore, are a source of variation in the final analysis. It is advisable to test the body alone, but the body structure is such that localized distortion of the body panels causes the test results to be rather inconsistent. Tests of the body alone are usually conducted to investigate the strength or rigidity of some component, and any test setup will be determined by the kind of information the engineer desires.

Torque is a turning moment or tangential effort. When torque is applied to one end of a unit being tested and some other portion is held securely, the unit is subjected to *torsion.* For a body-frame combination, the unit is supported usually at the front- and rear-axle locations, and deflection indicators are placed along the frame side rails and rocker panels (lower edge of body) at significant points where rigidity is to be determined (Fig. 1). The torque load may be applied to one side of the frame side rail through the lower control arm of the blocked out front suspension at the centerline of the front axle on the lower pivot pin by means of a hydraulic jack. The opposite side of the frame can be mounted on a mechanical or a hydraulic jack, depending on whether or not concurrent rotation of both sides of the unit or raising and lowering of one side only is desired. Either method is reliable and accurate, but the latter saves time through ease of testing.

The calculations to determine torsional rigidity of the unit tested depend upon the torque load applied and the angle through which the unit twisted as a result of the load. As an example, suppose that the average linear deflection measured at the front axle was

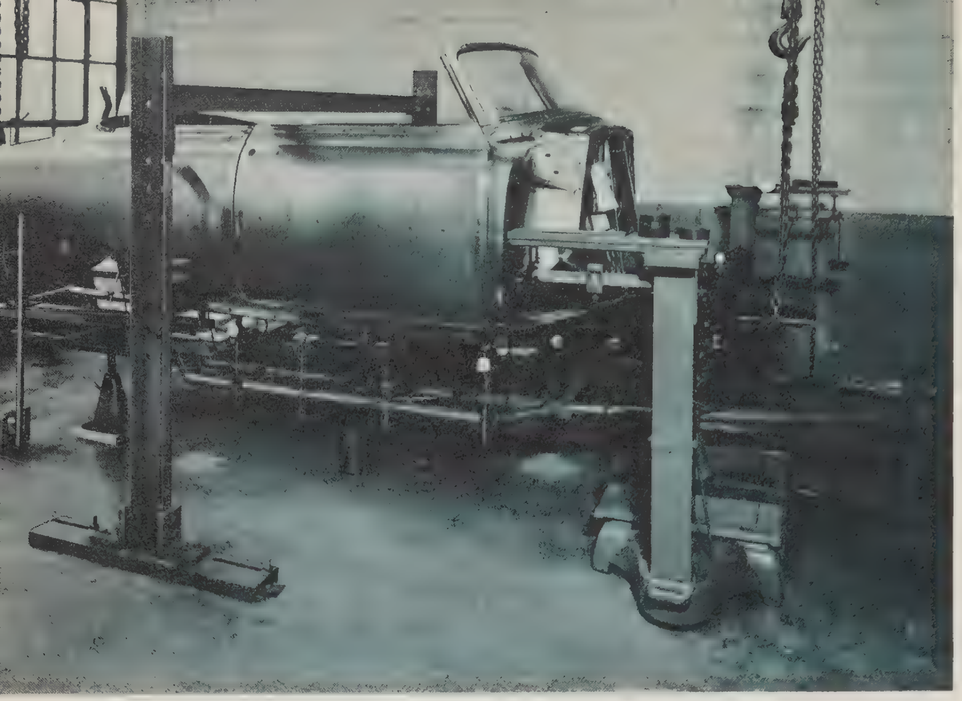

Fig. 3—The vertical bending characteristics are determined in this test setup of a body-frame combination. The method of load application is to impose a static load of 1,500 lb at the seat locations—750 lb on each seat. The deflection is recorded by dial indicators located along the underside of the frame rail and along the body rocker panels. A greater number of indicators is used to ascertain whether localized distortions are taking place. Bending test setups are identical regardless of body style. The support conditions vary, however, depending upon the type of rear spring suspension. The bending test indicates the automobile's ability to resist vertical deflections, such as those resulting from both front wheels hitting a road irregularity at the same time.

0.206 in. at a lateral distance on the frame of 31.50 in. For such a small deflection, the linear movement may be considered as an arc length measured in radians; therefore,

$$\phi = 57.3/LD$$

where

ϕ denotes angle of twist (deg)
L denotes lateral distance between indicators (in.)
D denotes average linear deflection (in.).

Substituting numerical test values:

$$\phi = 57.3/31.5 \times 0.206 = 0.375 \text{ deg.}$$

It also is recognized that this calculation can be eliminated by mounting the indicators 57.3 in. apart at stations where the deflection is required. Special gage support bars are required when deflections are recorded in this manner. These bars extend across the unit with the ends equidistant from the longitudinal centerline of the unit. The bars must be attached to the unit in such a manner that they will not reinforce the structure being tested.

The applied torque, considering a couple at the point of load application, is equal to the magnitude of the load applied on one side, multiplied by the arm of the couple. The arm is the lateral distance between the jacks:

$$T = FD$$

where

T denotes torque (lb-ft)
F denotes load—one side—(lb)
D denotes arm of couple (ft).

Therefore, assuming 2.5 ft between jacks and a load of 600 lb:

$$T = 600 \times 2.5 = 1{,}500 \text{ lb-ft.}$$

Knowing the torque applied and the angle of twist of the unit, it is evident that the resistance of the unit to twisting, *torsional rigidity*, R_T is the applied torque divided by the angular twist. Therefore, $R_T = 1{,}500 \text{ lb}/0.375 = 4{,}000$ lb-ft per deg.

It is convenient to assume an identical applied torque when plotting the twist of all the units tested in a comparison. The curves in Fig. 2 are plotted from a load of 1,000 lb-ft using the rear axle as the origin. Torsional rigidity constants, other than the front-to-rear-axle value, may be computed from values deter-

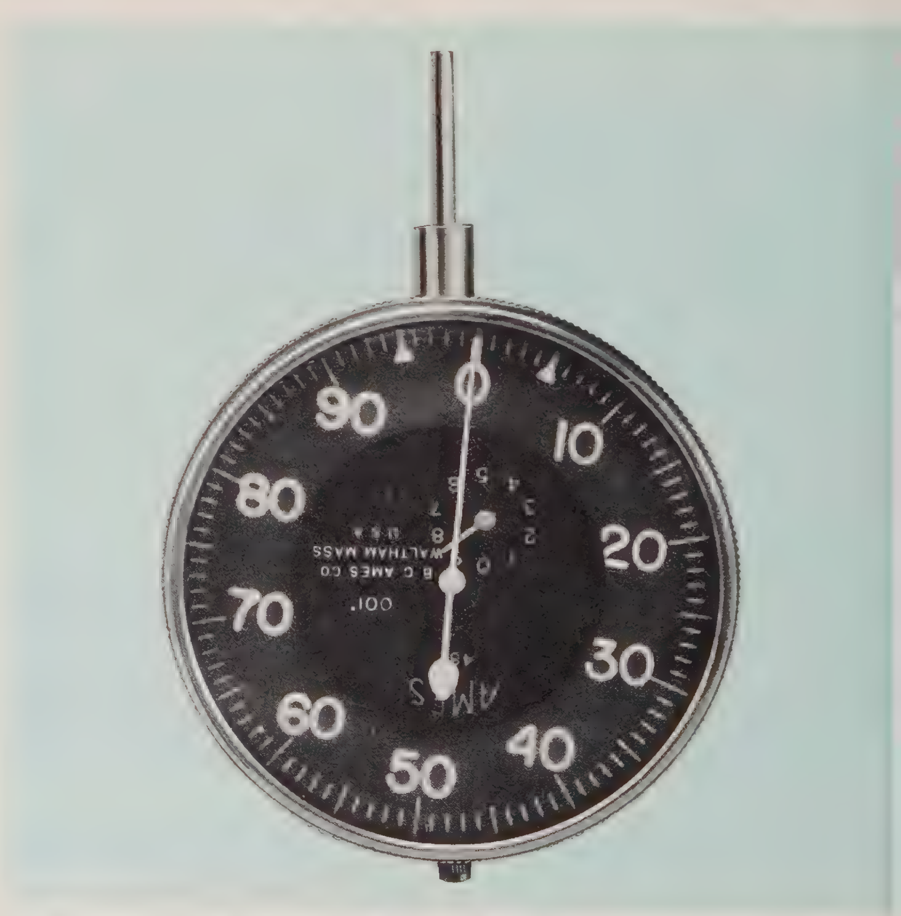

Fig. 4—Indicators of this type measure the deflection in inches during body structural tests. Mounting is arranged as shown in Figs. 1 and 3.

mined from the curve or actual deflections obtained at intermediate stations during the static torsion test.

When comparing cars of different wheelbase, it may be convenient to express the rigidity of the tested structure in terms of an average torsional stiffness per foot of length. This particular constant can be determined by multiplying the overall rigidity constant by the wheelbase in feet and obtaining thereby the average number of lb-ft of torque required to twist a 1-ft length of car through an angle of 1°.

The Vertical-Bend Test

While the torsion test is concerned with determining the ability of the structure to resist twisting moments (such as those induced when one front wheel hits a bump), the bend test is designed to resist vertical deflections such as those resulting from both front wheels hitting a road irregularity at the same time. Both conditions must meet definite standards or, as stated previously, the effects on the complete car will be quite pronounced.

The setup for the vertical-bend test differs from the torsion test in the method of load application. The type of rear-spring suspension must also be considered in order that the load distribution to the unit will compare as nearly as possible to actual road conditions.

For leaf-spring rear suspension, the frame or body-frame combination is supported at the rear-spring front and rear hangers. A channel iron or other stiff member is substituted for the springs and connected to the front-spring hanger. The rear-spring hanger rests upon the channel-iron beam and is free to move. The channel iron or member used is supported on a knife edge at the centerline of rear-axle location. The front end of the unit is supported on two jacks at the centerline of the front axle of the front suspension lower control arm with the suspension blocked out. For coil-spring rear suspension, the support conditions are the same as in the torsion test.

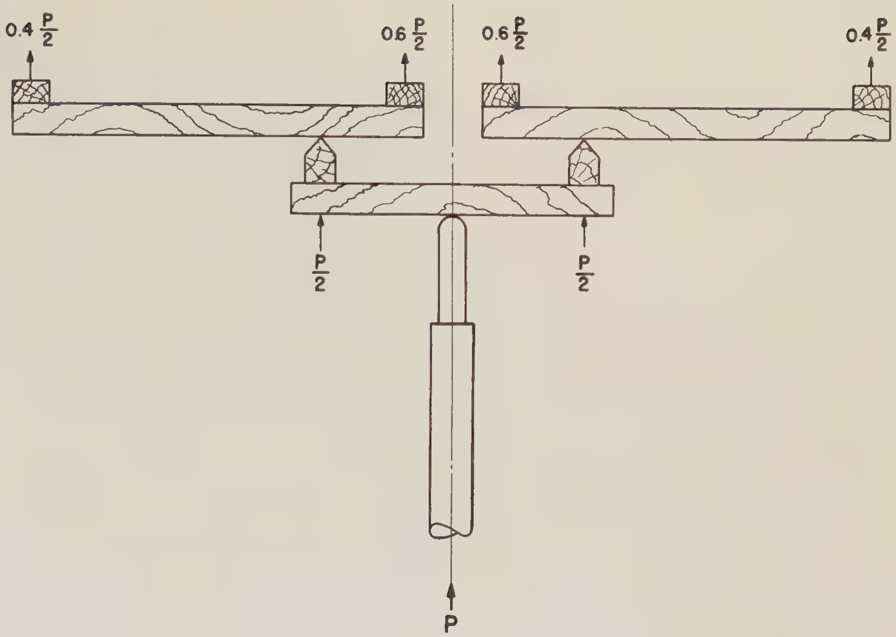

Fig. 5—In the seat frame test to determine the deflection and permanent set of a proposed design, a load P is applied by means of a hydraulic jack to a two-door sedan seat back as indicated above. This diagram illustrates that the two side members at the center of the seat back receive 60 per cent of the applied load. This figure is accepted as an empirical factor for the load distribution when three persons are seated.

The static load which is applied to the body is imposed at the seat locations. It has been determined that this method of loading produces a curve which more closely approaches the reaction which is received on the road. A load of 1,500 lb is considered a fair representation of its expected static load. If deflection due to higher or lower loads is desired, then a mathematical proportion, based on the fact that the deflection varies directly as the load applied, may be solved using the deflection resulting from the 1,500-lb load.

The 1,500-lb load is applied to the unit at the seat locations—750 lb at the front seat and 750 lb at the rear seat. The deflection is recorded by dial indicators located along the underside of the frame side rail and along the body rocker panels (Fig. 3). A typical indicator is shown in Fig. 4.

The calculations for the bend test are not as involved as for the torsion test. In applying the load to the body, the loading cycle is repeated several times from no load to maximum. The readings obtained from each cycle are averaged for each station along the length of the unit to obtain the average deflection of the body-frame combination at maximum load.

The result obtained is plotted as a preliminary curve. Any deflection obtained at the front-axle and rear-axle locations is considered as *support* deflection. By joining the deflection points at the supports with a straight line and re-plotting the curve with the straight line representing zero support deflection, the curve assumes its final form (Fig. 2).

The curves obtained from both the bend and torsion test are useful to the designer in locating areas of localized deflection. The load distribution to the rocker panels can be ascertained quite readily from a plot of the test results. Chassis frame outrigger stiffness as well as the underbody cross-member structure can be checked with ease through the use of the vertical-bend test.

Jacking Test

The jacking test is conducted primarily to determine any excessive door or deck-lid opening distortion. When a motorist experiences a flat tire, he usually jacks the car at the bumper near the wheel in question, thereby loading the body at that point. If distortion of the body openings is excessive, it may cause binding of the door or deck-lid panels, as well as the locks.

In an attempt to prevent such difficulties, a jacking test is conducted on the body-frame combination. The method of supporting the unit is identical to that in the vertical-bend test. Jacking is performed at the extreme ends of the frame side rails, front and rear. Indicators are mounted off the door to record movement of the doors relative to their lock pillars, and other indicators are mounted to record movement along the vertical centerline of the body and horizontal

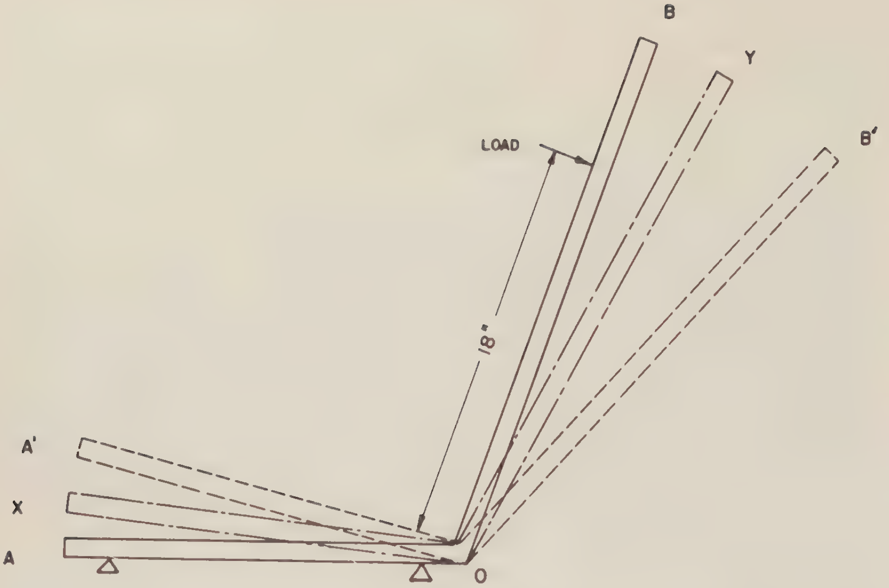

Fig. 6—This schematic diagram illustrates the various conditions of deflection and permanent set of the seat frame structure. Standard design practice accepts a reasonable amount of deflection and permanent set. Position $A'OB'$ denotes the seat under load. The seat returns to position XOY when the load is released. BB' is the maximum deflection of the seat back; BY is the maximum permanent set. While deflecting the seat back, however, the seat bottom rotates through the distance AA' and returns to a permanent set AX.

centerline of the deck-lid opening.

In jacking a body-frame combination, sufficient dead weight should be added in the body and on the chassis frame to equal the weight of the engine, seats, instruments, and other car equipment.

Analyzing jacking test data is simply a matter of comparing the maximum deflections obtained and their direction of movement with tolerances specified on the product drawing. It is obvious that if the movement exceeds the allowed tolerance, binding will occur. It is usual practice to observe the closing action of the doors and rear-compartment lid under maximum jacking load and to report any deficiencies in need of correction.

Shake-Rig Tests

The tests just described are conducted on the completed body prior to painting and trimming. These tests present an overall picture of the structural characteristics of the body from a static standpoint.

While these tests require no special equipment, another test uses a machine known as a *shake rig*. This machine was developed by the Fisher Body Division in attempting to verify further the structural characteristics of the body. The shake-rig testing of a body or body-frame combination is more a visual study of the body than an actual test. After a unit has been mounted and the machine set in motion, the action of the body members as it might occur on the road can be observed.

The value of the study can be recognized from a description of the shake-rig machine.

The shake rig consists of two platforms —the lower platform or base which always remains stationary, and the upper platform which is activated by cam motion. Two cams are located on each side of the shake-rig base and are linked to the upper platform. By alternating the cams (setting cams on one side 180° out of phase with its opposite), a rotary oscillating motion of the upper platform results. When all cams are in phase (parallel), the upper platform remains level, and moves in a vertical direction. The cam settings can be adjusted for various displacements which then remain constant. By varying the speed of the shake rig, the action of the body under different road conditions may be analyzed in the laboratory. The shake rig affords the engineer an opportunity to try various designs in order to bring the body up to standard or to eliminate unnecessary members. Changes of this kind can be accomplished while the body is in motion.

Seat-Frame Tests

Front seat structures do not contribute to the structural characteristics of the body. However, their importance to the user of the vehicle demands an extensive test program to determine their independent structural strength and stiffness. Consequently, seat tests are an important part of any body-test program.

The seat adjusters and mounting brackets, after having been tested and found satisfactory, are attached to the seat-bottom structure for the entire test program. The seat-bottom structure is loaded vertically to make certain that it will not collapse when subjected to excessive loads. The seat-bottom rear crossbar, a very critical member of the divided back seat, must be sufficiently rigid to resist permanent set caused by excessive loads on the seat backs. After undergoing a series of tests, the seat is carefully examined to determine any critical failures, such as broken welds, local buckling and possible sources of rattles, or other seat noises. The seat structure first must be approved before the installation of springs and cushions.

Loads can be applied to the seat backs to determine their deflection rates and permanent set or distortion after loading. The method of determining seat-back deflection for both four-door sedan and two-door sedan structures is identical, with the exception of loading. The load is applied to the seat-back structure of both designs. However, the four-door sedan seat back is loaded simultaneously through each of its two side members. A beam is placed across the width of the seat back approximately 18 in. up from the rear-seat adjuster bolt and secured to the side members. The load is applied to the center of the beam by means of a hydraulic jack, thus distributing one-half the applied load to each side member.

The two-door sedan seat back is in two sections, each section having two side members. In the beam setup for loading a divided back seat, the two side braces at the center of the seat back receive 60 per cent of the applied load (Fig. 5). The 60 per cent has been accepted as an empirical factor for the load distribution when three persons are seated.

In order that a seat can be considered satisfactory, it must withstand a load of 1,000 lb. A reasonable amount of deflection and set of the seat structure is expected, however, under standard design practice. The seat-back deflection is the measured deflection of the seat-back upper cross member minus the

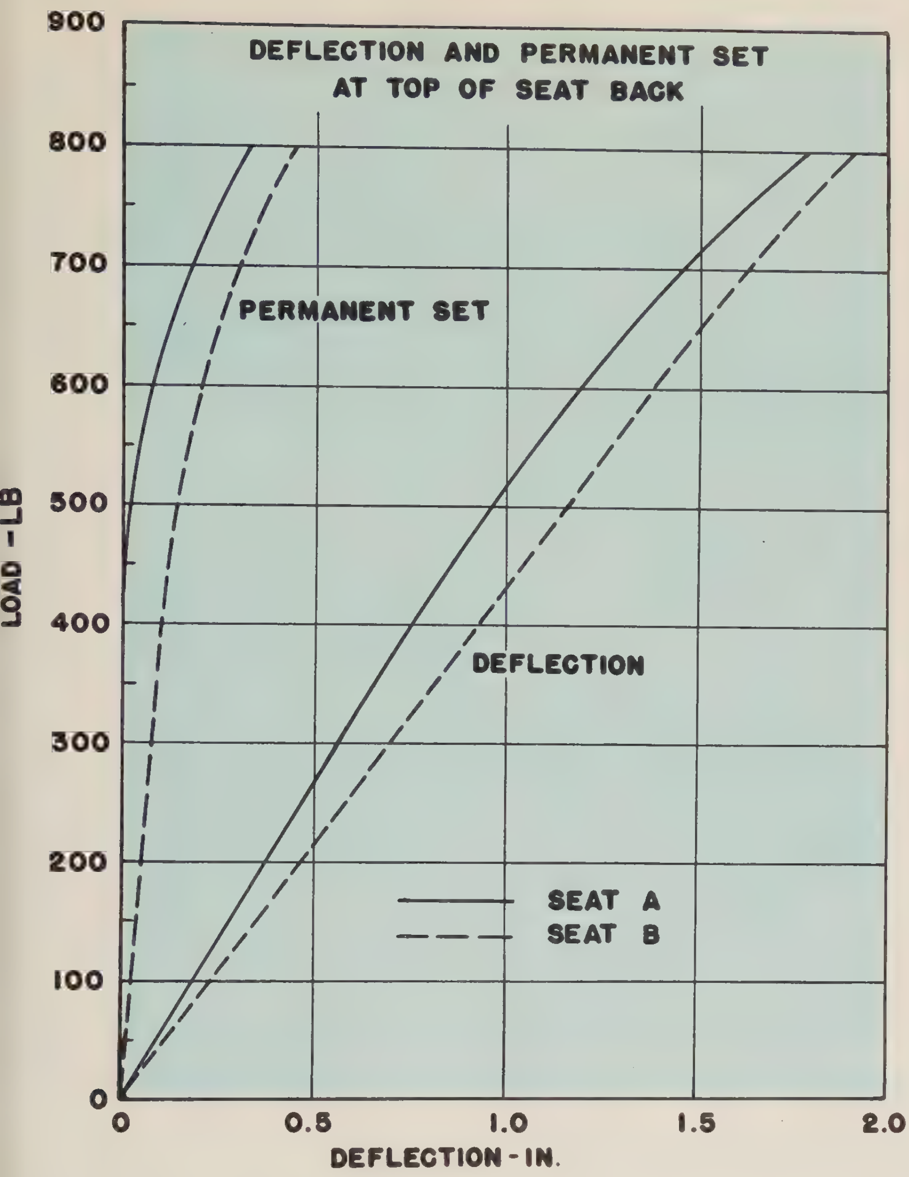

Fig. 7—The deflection and permanent set data obtained from a seat back test are plotted for comparison with previously determined standards. In the above curve, the proposed seat *A* compared favorably with seat *B* used as a standard.

deflection resulting from rotation of the seat-bottom structure.

To determine whether the amount of deflection and set in a proposed design will be acceptable, certain calculations must be made from the test data. Fig. 6 is a sketch representing a seat structure. The seat under load assumes position $A'OB'$ and, when the load is released, returns to position XOY. Horizontal deflection at point O is generally slight in comparison with the maximum deflection and can be neglected without any appreciable error. The deflection BB' is the maximum deflection of the seat back, and BY is the maximum permanent set. However, while deflecting the seat back, the seat bottom rotates through the distance AA' and returns to a permanent set AX.

To determine the net seat-back deflection D_N, the rotation of the seat bottom is considered as support deflection and must be deducted from the total seat-back deflection D_T. Therefore, correcting for the differences in beam length between the seat back and seat bottom:

$$D_T = BB' - \left(\frac{OB}{OA} \times AA'\right).$$

The net set S_N must then be deducted from the total seat deflection D_T. The net set is calculated as follows:

$$S_N = BY - \left(\frac{OB}{OA} \times AX\right).$$

Thus, to obtain the net deflection of the seat back D_N:

$$D_N = D_T - S_N.$$

It is general practice to make the computation and plot the results in the form of a rate curve to study the bending characteristics of the seat back in comparison with other seats. In Fig. 7, typical deflection and permanent set curves for a proposed coupe seat, seat *A*, are compared with a standard seat, seat *B*. These curves show that seat *A* is a satisfactory structure when compared with the standard, seat *B*.

Summary

The advantages to be gained by a well-planned series of laboratory tests are exemplified by the major tests discussed herein. These tests have been selected because they have proved their worth in the past and are presently being used to better the quality of future models.

Laboratory tests to determine the torsional and bending rigidity of a body are conducted according to certain fundamental engineering principles. The torsion- and vertical-bend tests definitely establish whether or not the body has sufficient stiffness to perform as capably as models of the past. The jacking test is further proof that the body will withstand the severe use to which it may be put by the customer.

In addition to the major tests, it is essential that each load-carrying member of the body structure be tested to determine if the part is dependable functionally and is designed on the basis of sound engineering principles. The seat-structure test is a fine example. The necessity of obtaining a seat structure which will insure long cushion life and, obviously, greater passenger comfort is a vital part of the Fisher Body test program.

Final proof that the automobile body is ready for production is only obtained through road testing. Road tests, or complete car tests, are followed closely by the test engineer so that he may be in a better position to correlate future laboratory results with road-test results as well as to obtain verification of the tests already conducted in the laboratory.

Analyzing a Typical, Simple-Cycle Turbo-Jet Design and Development Program

Since the end of World War II the turbo-jet engine has undergone a period of intense and accelerated development. Hand in hand with the accelerated development program have come the problems of insufficient design study time and lack of production facilities. The comparative newness of the turbo-jet engine has contributed to a shortage of trained personnel capable of entering and moving along with a stepped-up engine design program. Accelerated schedules tend to exaggerate the importance of particular phases of jet engine design, leading to the de-emphasis of other steps. In order that the amount of time available for the development of an engine be utilized to the fullest extent, the engineer should have available a step-by-step procedure or timetable, and all of the design personnel involved should thoroughly understand the position and importance of their own particular design phase in relation to the broad overall picture of jet engine development.

All of the thermodynamic, mechanical, control, and liaison aspects of turbo-jet design are investigated for their individual importance and effect upon the engine design as a whole in order to determine and evaluate the major steps of a jet engine design program. The major basic design procedure steps for a typical simple-cycle (non-afterburning) turbo-jet design and development program are as follows:

1. Establish the need for a specific jet engine.
2. Determine by cycle analysis the desired engine operating conditions to meet the specific need.
3. By use of performance calculations check the performance of the cycle-analysis-choice engine over the entire range of expected engine operating conditions.
4. Correlate the need, cycle, space, and performance characteristics of the selected engine to see if it will meet the specific requirements needed. If not, repeat steps 2 and 3 until a proper choice is made.
5. Examine the possible compressor and turbine blading designs to ascertain if it is feasible to construct the selected engine. If so, select the compressor and turbine blading designs to be used in the engine.
6. Establish a preliminary layout of the selected engine. This preliminary layout may indicate that it will be difficult to construct the engine as originally conceived, and a new basic design must be selected. If possible, several different layouts should be made in order to compare ideas and methods. A little extra time spent in this stage of engine design may save considerable re-design time in the later stages of development.
7. Make a thorough stress analysis of the preliminary layout.
8. Select a suitable oil system configuration based on the preliminary layout.
9. Calculate the weight of the preliminary layout engine. Excessive weight may dictate extensive re-design.

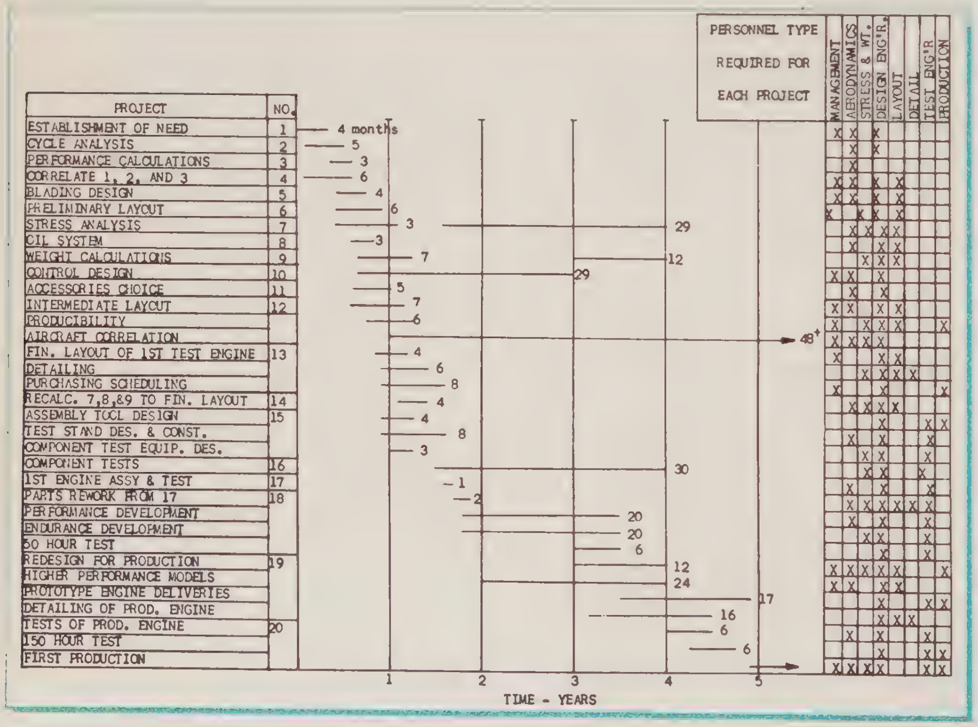

Fig. 1—This is a typical five-year development schedule set up for a simple-cycle (non-afterburning) jet engine. To the right of each major step in this particular program the time allotted is indicated by the marked area. At the right-hand side of the chart are shown the types of personnel required by each development step or stage. The time-allotted areas clearly show how many of the steps overlap one another and show the complexity of the jet engine development program as a whole.

By MERRILL G. HINTON, JR.
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A design program can be complicated; but it also can be organized for efficiency

10. Select suitable fuel and exhaust area control systems.
11. Select the necessary accessories such as fuel pumps, starter, and hydraulic pumps, as required by the preliminary layout.
12. During a period of intermediate layout establish the external configuration of the engine, such as controls and accessories piping and location, in addition to studying the producibility of the various engine components. Extensive correlation with the aircraft manufacturer is necessary in determining the external engine configuration.
13. Make a final layout of the first test engine, utilizing the best features found during the periods of preliminary and intermediate layout. All parts must be detailed and purchasing schedules set up.
14. Recalculate the stresses, weight, bearing loads, and oil system requirements for the final layout engine.
15. Design equipment to assemble and test the engine and its components.
16. Perform component tests.
17. Assemble and test the first test engine.
18. Complete performance and endurance tests on the final layout engine culminating in passage of the 50-hr preliminary flight rating test.
19. Re-design the fully tested final layout or prototype engine on the basis of production requirements, if necessary. Deliver prototype engines to the airframe manufacturer.

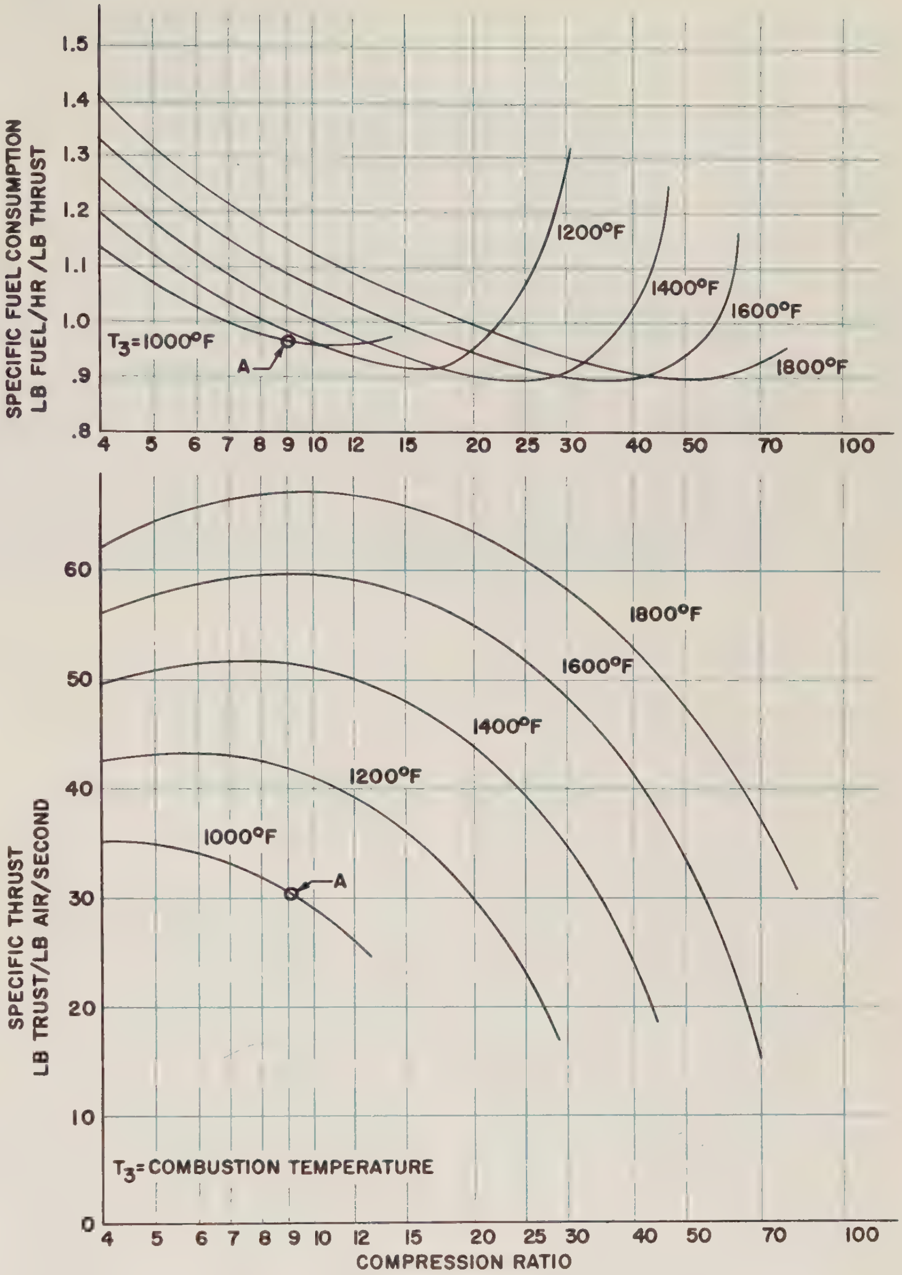


Fig. 2—From these curves the specific fuel consumption *SFC* and the specific thrust $F_n W_a$ for various compression ratios R_c and combustion temperatures T_3 are clearly indicated. A condition of operation is selected at a feasible R_c with a reasonable T_3 and good *SFC*. The total engine air flow is determined by dividing the thrust required by the aircraft by the specific thrust of the engine. It should be noted that the cycle analysis is made for one altitude only, as determined by the design point, and this selection might be an engine which could not meet the aircraft requirements at operating conditions other than the design point. For example, an engine chosen for condition *A* of the cycle analysis curves in this illustration (minimum *SFC* at R_c of 9 to 1 at 35,000-ft altitude) might be unable to deliver the proper take-off thrust at sea level, although it could develop the proper cruise thrust at altitude. Therefore, non-dimensional performance curves are developed which evaulate the performance of the selected engine over the wide range of expected operating conditions. In developing the performance curves for the engine, efficiencies for the various components of the engine (for instance, compressor, turbine, diffuser, and exhaust nozzle) are assumed and the flow and pressure characteristics throughout the complete engine are estimated.

20. Test the production design, resulting in completion of the 150-hr qualification test, clearing the engine for its initial goal—production.

It is obvious that some of these steps overlap; that is, one step cannot be completed before the next step is begun. That is one of the major problems of the

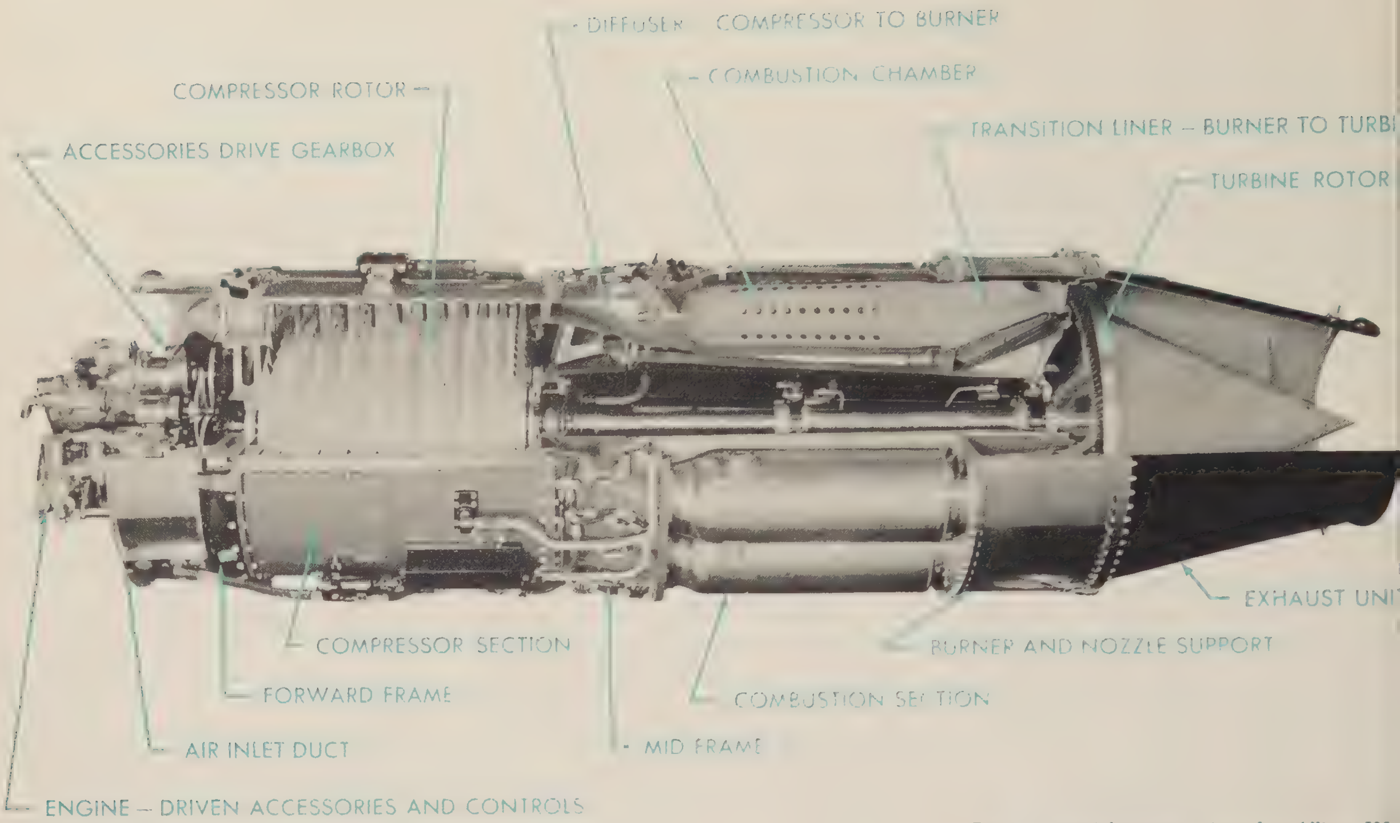

DIFFUSER — COMPRESSOR TO BURNER
COMBUSTION CHAMBER
COMPRESSOR ROTOR
ACCESSORIES DRIVE GEARBOX
TRANSITION LINER — BURNER TO TURBI
TURBINE ROTOR
EXHAUST UNI
BURNER AND NOZZLE SUPPORT
COMPRESSOR SECTION
FORWARD FRAME
COMBUSTION SECTION
AIR INLET DUCT
MID FRAME
ENGINE — DRIVEN ACCESSORIES AND CONTROLS

Fig. 3—A partial cutaway view of an Allison J35 turbo-jet engine showing the more important aerodynamic and structural features of the engine.

engineer—how to allocate the necessary personnel and time to each step of the development program and still keep on schedule (Fig. 1).

Period of Establishment of a Need

In establishing the need for a specific jet engine the engine manufacturer holds many conferences with officials of the Air Materiel Command (A.M.C.) of the United States Air Force (U.S.A.F.) and the Bureau of Aeronautics (BuAer) of the United States Navy (U.S.N.). These agencies perform the long-range and immediate planning of the U.S.A.F. and U.S.N. aircraft requirements for defense purposes. The A.M.C. and BuAer award the development and production contracts to the various engine and aircraft manufacturers.

In discharging their responsibilities the A.M.C. and BuAer hold design competitions for each model aircraft they deem necessary. In these design competitions the A.M.C. and BuAer merely stipulate the general features of the aircraft, such as payload, range, take-off distance, landing distances, and maximum cruise speeds, that are considered essential. It is then the task of the aircraft manufacturers to design the airframe and determine the requirements of the engines necessary to supply the propulsion power. A.M.C. and BuAer name a number of available engines that the designers can choose from, but it is possible that none of the current engines is suitable for the newly designed aircraft. This condition establishes the need for a new model engine.

It is the responsibility of the engine manufacturer seeking to develop a new engine model to propose new models which would be suitable for the new aircraft models under consideration or needed by the military services. In making the proposals for new engine models the engine manufacturer goes through a period of cycle analysis to determine the basic engine size and operating characteristics. These cycle-analysis calculations result in a family of curves as shown in Fig. 2.

If the preliminary engine choice does not meet all of the required operating conditions of the particular aircraft or assumed need, then another basic design point is chosen from the developed cycle-analysis curves and the jet engine performance re-evaluated for the new basic design point. When an engine choice is selected that meets the aircraft requirements under the necessary operating conditions, then the performance parts of the engine are thoroughly analyzed to determine whether or not it is mechanically and aerodynamically feasible to construct the performance part in the space allotted to the engine by the airframe *nacelle* (engine housing) size.

In this period of establishment of a specific need, then, members of management confer with military officials and aircraft manufacturers. Aerodynamicists perform cycle-analysis and performance-analysis functions, and design engineers evaluate the preliminary physical features of the proposed new turbo-jet engine.

Preliminary Aerodynamic and Mechanical Design Period

In conjunction with and as a result of the basic engine cycle choice from cycle-analysis and preliminary performance calculations, the blading design for the compressor and turbine is analyzed to ascertain the feasibility of constructing these basic engine components in the allocated spaces to have the predicted performance characteristics. A choice is made as to the type of flow pattern that will be used in the compressor: free-vortex, constant reaction, half-vortex, or solid rotation. The basic type of compressor is decided upon: subsonic, transonic, or supersonic. The distribution of pressure rise, temperature rise, and air velocity through the compressor is worked out, based upon expected stage efficiencies and limiting flow *Mach numbers*. (Mach number is equal to the velocity of a body divided by the speed of sound in the atmosphere through which it is moving.) After the basic velocity or work diagrams are constructed for the compressor the details of the compressor blades, including such features as solidity, setting angles, camber angles, and incidence angles, are calculated in order to assure the attainment of the velocity diagram indicated performance.

The turbine functions as the unit which supplies the necessary power to drive the compressor and various accessory units and, as such, its basic design is determined primarily by the compressor power requirements. As in the case of compressor blading design, the flow system that will be used for the turbine is chosen: free-vortex, constant reaction, or one of the others mentioned previously. The type of turbine blading, impulse or reaction, and the work distribution or number of turbine stages is chosen, followed by calculations of the detailed design features of the blades.

The preliminary layout of the engine establishes the physical configuration of all the major engine components and is based primarily upon the compressor and turbine blading designs, with the combustor design being of secondary importance. The preliminary layout is used as a framework, a model upon which to base estimates and calculations for special features, controls, and accessories.

At this point, the designer has available:

- Compressor length, effective outside diameter, blading configuration, and number of stages
- Turbine length, effective outside diameter, blading configuration, and number of stages.

Together with this information, the necessary space requirements for mixing and burning of the fuel and air are calculated. In general, this feature is approximated from data and tests of combustion in passages of varying size, shape, and capacity. More often than not, the information gained from experience with a previous model engine is a deciding factor, unless the proposed combustion chamber is of a new design.

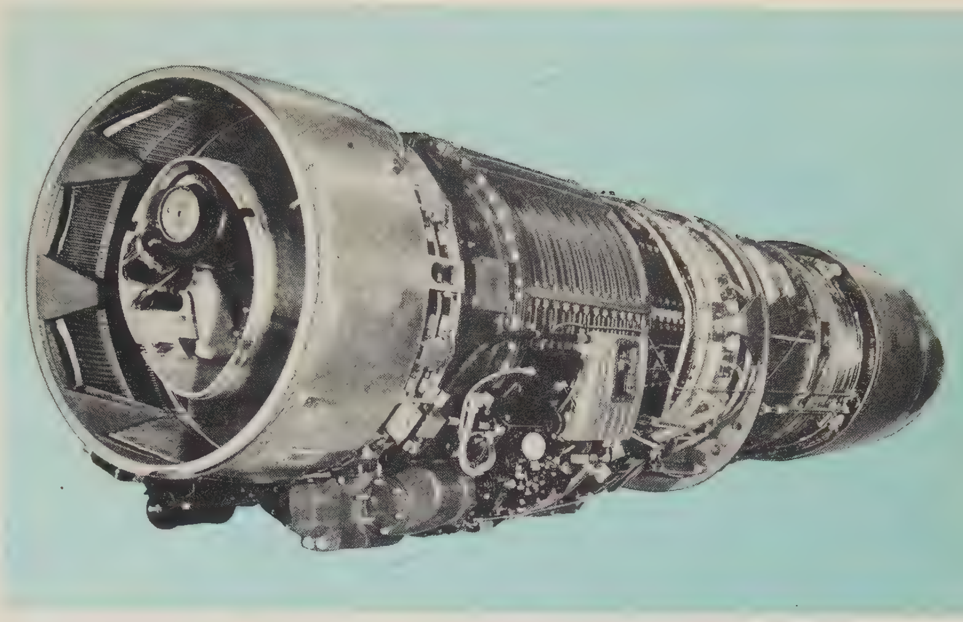


Fig. 4—Exterior view of Allison J71 model turbo-jet engine showing the various control and accessory components mounted on the outer shell and inlet to the engine.

Provision is made for inlet to the compressor, diffusion from the compressor outlet to the burner inlet, transition from the burner outlet to the turbine inlet, and expansion from the turbine to ambient surroundings. These features establish the flow passages of the jet engine and are influenced by such flow factors as passage shape, area, streamlines, and fluid velocity and pressure. The flow passages are designed to keep the pressure loss through them at a minimum. This fact is contradictory to engine length limitations since smooth-flow cross sectional areas without rapid changes in contour are essential for a low pressure loss design, while every effort must be made to keep the engine length and weight at a minimum for aircraft installation and operation considerations. Therefore, a balance is reached to give the best overall effect. The diffuser passage from the compressor exit to the burner inlet is very critical in that it determines the velocity profile and axial air velocity entering the burner, which is an important consideration in the combustion efficiency and flame stabilization characteristics of the combustion chamber.

With the high pressures employed in the present gas turbines, the resultant compressor and turbine thrust bearing loads due to gas pressure acting on the faces of the compressor and turbine blades and the end wheels may become excessively high. To alleviate this condition, labyrinthian sealing of the gas passages through the engine is used. In addition, compressor bleed-off air is used in dummy piston areas and coupling of the compressor and turbine rotors is given special design emphasis.

The arrangement of the main engine bearings for the compressor and turbine shafts, the location of the mounting connections between the engine and the airplane nacelle or fuselage, and the type and location of accessories drive gear box are considered.

The engine basic shell structure, including forward frame, compressor cases, mid frame or diffuser casing, combustion chamber case, burner support or transition casing, and turbine cases, is designed to satisfy the engine mount load requirements, pressure forces, and flight maneuver forces.

An exhaust unit, whether a fixed nozzle or variable area nozzle, is designed. Also, an air inlet duct, which may or may not include retractable air inlet screens and doors, is included.

At this point, the mechanical details of the basic layout are finalized. These details include: (a) shaft, wheel, and blade design for compressor and turbine rotors, (b) compressor stator configuration. (c) integration of a power take-off gear box in either the forward frame or mid frame, (d) selection of bearing and gear arrangement for accessories drive gear box, (e) structural design of bearing support structures such as mid frame, forward frame, and burner support, (f) inner and outer case structure for combustion section, (g) bleed-air cooling arrangements for critical items such as turbine nozzles and wheels, (h) bleed-air arrangements for control of compressor surge during engine acceleration, if required.

The oil system necessary to lubricate the bearings and gears is added to the layout. This required an estimation of bearing and gear requirements, pressure losses, heat transfer, and pump capacities.

In conjunction with and hand in hand with the layout, each component part is thoroughly stress analyzed to make certain that none of the basic units is designed in such a manner as to cause or to receive excessive loads or stresses. At the same time the weight, mass moments of inertia, and critical material content or selection are established. Stress analysis is used to help pick the proper material for use in each engine part, enabling the designer to use the lightest weight part possible to do the job—a critical factor in jet engine design. The importance of proper stress analysis of the entire engine can be emphasized by the fact that a single small blade retaining pin, if improperly designed, could permit a turbine or compressor blade to fly loose and perhaps ruin the entire engine.

The preliminary engine layout period, consisting of basic layout and mechanical details, is a somewhat long and complicated process utilizing the combined efforts of management, aerodynamics, stress, design, and layout personnel. At the end of the preliminary layout a complete picture of the basic engine is available, from which the necessary controls and accessories are selected. Fig. 3 is a partial cutaway view of a turbo-jet engine showing the various engine components.

Controls and Accessories Selection

The turbo-jet engine speed and power output are a direct function of the amount of fuel burned for any given engine operating at a given altitude and flight Mach number. Thus, a control unit is selected to control the fuel flow to the combustion chamber which, in turn, controls the engine speed and power or thrust output. The choice of control actuation is selected from the basic mechanical, hydraulic, or electronic configurations, or combinations of these systems. In addition, if a variable area jet nozzle is incorporated in the engine design, a method of controlling the jet nozzle actuating mechanism is devised. A convenient arrangement is to have the engine fuel control mechanism and variable area jet nozzle control mechanism coordinated.

Other fuel system components such as fuel pumps, filters, nozzles, and lines are selected for the specific engine capacity. Tanks, lines, valves, actuators, and other components are designed for the hydraulic actuating system of the engine. A starter, tach generator, alternator, thermocouples, and miscellaneous valves, lines, fittings, and bosses are selected to complete the basic engine accessories and control circuits. In the process of selecting controls and accessories, management, aerodynamicists, and design engineers work together to assure that the engine's needs are satisfied.

Intermediate Engine Design Period

The process or period of intermediate engine design has to do with the establishment of the external physical configuration of the engine and the design of special features such as anti-icing, air inlet screens and doors, and variable-area tailcone. It should be realized that the jet engine is being designed, in each case, for specific aircraft installations. Therefore, engine manufacturer liaison groups have extensive correlation contacts with the aircraft manufacturer to establish the overall physical size and configuration. Agreement is reached upon the placement of the accessories, the size and routing of the piping, control cables, and rigging. One very important item is the agreement as to how the airframe nacelle or fuselage mounting system shall support the engine. A poorly designed airframe support structure can impose excessively high mount loads upon the engine structure. A method is jointly agreed upon to protect the airframe from the temperature effects of the engine. The engine manufacturer establishes the limitations of engine operating exhaust gas temperature and aircraft flight speed. It should be emphasized that correlation between the engine and aircraft manufacturers is not a one day or one month job. It begins when the preliminary facts about the engine physical configuration are first available and continues over the whole period of engine development.

In this intermediate period the final layout of the engine is selected and detail drawings are made of the individual parts. Upon completion of the detail drawings, purchasing schedules are set up to insure on-time delivery of all parts.

At the same time, the stress, weight, bearing load, and oil system requirements of the final layout engine are reviewed in the light of any changes made over the preliminary engine layout. Also assembly tools are designed, test stands built or revamped, and component test rigs designed and built.

In this period of intermediate design, the full complement of the engineering department, management, aerodynamicists, stress and weight analysts, design engineers, layout men, detailers, and test engineers works in full cooperative action in order to complete the design work on schedule. Fig. 4 illustrates a fully designed turbo-jet engine built to the final engine layout.

Component and Engine Testing

As previously mentioned, special equipment has been designed and built to give static, endurance, and control tests to the various engine components and accessories. These tests begin as soon as sufficient parts are received for this program. The results of these tests supplement the original design assumptions with actual data under simulated working conditions. Often, the component tests indicate necessary design changes which otherwise would never be apparent until the engine is tested in the airplane. Fig. 5 shows an engine component part undergoing static tests.

After the engine design is finalized and test equipment procured, the engine(s) is (are) built and subjected to a series of tests which indicate both engine performance and durability. This test program is similar to the dynamometer testing of reciprocating engines and enables the design engineer to correlate final results with earlier ideas and assumptions. Performance tests of the engine enable the performance group to determine accurately the engine fuel flow and temperature requirements and make it possible to establish the settings and limits for the engine controls. Engine performance curves are recalculated on the basis of actual engine characteristics, rather than the assumed efficiencies used up to this point.

Hand in hand with performance development, the engine engineer schedules a series of endurance tests to indicate the degree of success of the engine mechanical design. This endurance testing program involves temperature surveys of the oil system, cooling air, engine skin, wheels, and blades. These temperature surveys indicate the actual running conditions that will be encountered and may differ greatly from those anticipated when the engine was initially conceived. The failure of any part due to rubbing, wear, cracks, corrosion, and scaling during the testing program, are definite indications of either poor design or wrong assumptions, and the endurance testing series gives the engine a chance to speak up and make its "ailments" known.

After considerable endurance and performance testing with subsequent design refinements to the engine, an engine is built and put through a 50-hr preliminary flight rating test. The 50-hr test must be completed before the airframe manufacturers can accept delivery of the engine model for use in their prototype aircraft. It entails a series of engine starts, full-power runs, cruise-power runs, accelerations, decelerations, and shutdowns, which total up to 50 hours of engine running time. The completion of such a test is considered to represent the completion of the major portion of the design and test work needed for the development of a new model jet engine. The way is then clear for design refinements and preparation for production.

In this period of component and engine testing, personnel from the aerodynamics, stress, design, drafting, and test groups compile and analyze engine performance and endurance characteristics.

Production Re-Design Period

After the completion of the 50-hr test the final layout engine is re-designed on the basis of quantity production requirements. At this stage in the development of the engine, the design engineers have available the results of extensive performance and endurance testing. They benefit from all this test experience and then combine the best mechanical and aerodynamic features presented with part detail designs which lend themselves to availability of raw material, procurement, lower weight, lower stresses, higher performance, simplicity of design for quantity production, and the absence of critical materials.

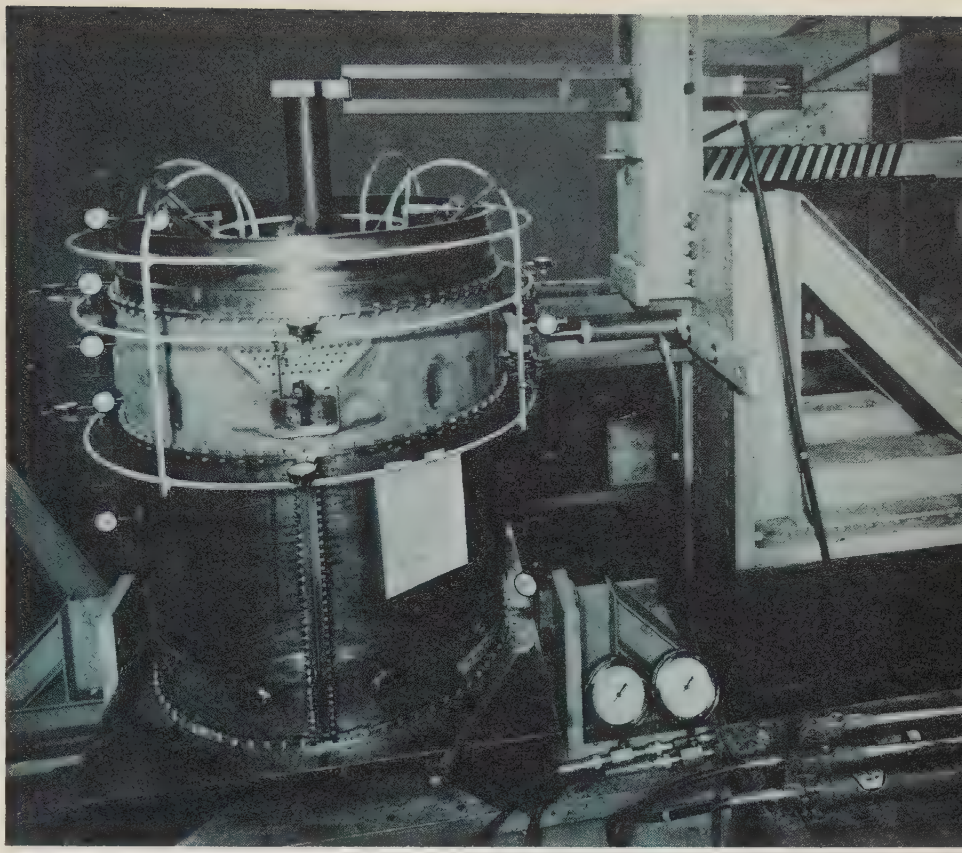


Fig. 5—A static-test laboratory setup showing a portion of Allison model J71 engine outer shell being loaded with simulated flight maneuver forces.

Upon completion of parts detailing for the production model engine, series of engine tests are run to determine the effect of any part re-designs or material modifications upon the engine performance and endurance, and the engine is subjected to the applicable military 150-hr qualification test, which the engine must pass before it is cleared by the sponsoring military organization for production. This test is similar to the 50-hr test mentioned previously in that it checks the engine's performance characteristics during starting, idle, accelerations, decelerations, normal-thrust, and maximum-thrust operations. The full complement of the engineering group, management, aerodynamics, stress, design, drafting, test, and production personnel cooperate fully in this period of production re-design.

Successful completion of the 150-hr test is a thorough proof of the engine's basic aerodynamic and mechanical design considerations and, as such, is a good and just reward for the engineers who labored with the engine from its initial conception on the drawing board, through various design difficulties and failures to the final goal—clearance for production.

Summary

The complexity of turbo-jet engine mechanical design, coupled with the numerous aerodynamic problems (compressor, turbine, combustion) and control requirements, necessitates that many different groups or activities be used to resolve these specialized tasks. The engine testing program and production requirements also require separate groups. Thus, it can be seen that the main overall problem is one of co-ordination of the activities of the various groups, of having the right information available at the proper time. A well planned design and development schedule is used to assure complete co-ordination and necessary progress for the entire turbo-jet engine development project.

Field Testing of Diesel Locomotive Axles

By LUDVIG PETERSEN and
ROBERT A. MOREAU
Electro-Motive
Division

How to know what the axle itself learns about stress conditions

To design a piece of equipment so that it is strong enough, but not unnecessarily strong, the designer must know what forces the equipment will be subjected to during its life. If this is not known, and it usually is not, then the engineer must replace this knowledge with assumptions and—to the extent of the intelligence behind these assumptions—he must either risk failure by underdesigning or be unduly safe by overdesigning, which results in unnecessary expense. In the case of Diesel locomotive axles, it was known from the way they stood up in service in the past that they were strong enough, but it was not known if they were so strong that their loads could safely be increased.

In view of this, Electro-Motive Division engineers embarked on a program of testing railroad axles in actual service under all the various conditions encountered in the field. A test of this nature offers many problems relating to instrumentation for the test and interpretation of the test data.

RAILROAD axles are used in large numbers and represent, therefore, a considerable investment. Railroad axles are among the most vital parts of a railroad locomotive or car because a failure of an axle can result in a very serious accident.

To a casual observer railroad axles may all appear very much alike, except for the size. Some have a little larger fillet radius than others and some a little larger wheel seat than others. There are other variations but they all seem rather minor. Fig. 1 shows a typical axle which is used by Electro-Motive Division for its Diesel locomotives.

This axle is made of high-carbon steel. It contains 0.40 per cent to 0.59 per cent carbon and 0.60 per cent to 0.90 per cent manganese. In addition, it contains small amounts of phosphorus (0.045 per cent max), sulphur (0.05 per cent max), and a minimum of 0.15 per cent silicon. After the axle has been heat treated and tempered, it has a tensile strength of 86,000 psi min and a yield point of 48,000 psi. It has a minimum of 24 per cent elongation in 2 in. and 38 per cent reduction of area before failure.

There is no legal code for the construction of railroad equipment similar to the Building Code for building construction. Except for certain safety features established by the Interstate Commerce Commission for Cars in Interstate Traffic, it is left to the railroad to insure that the equipment is safe for the life and property entrusted to them as carriers.

In view of this, it is no wonder that the railroads, primarily through their Association of American Railroads, have done a lot of work to learn about the necessary requirements for safe axle design. In this endeavor the railroads have enjoyed the cooperation of many universities and manufacturers. This has resulted in quite a bit of available information about railroad axles and especially about the good or bad effects of various details which have been tried out from time to time.

It is now possible to predict, with reasonable accuracy, what stresses will be induced in a specific axle design by a certain load and the relation between the stress and the number of stress applications which will cause a failure is also fairly well established. There is, however, very little information available

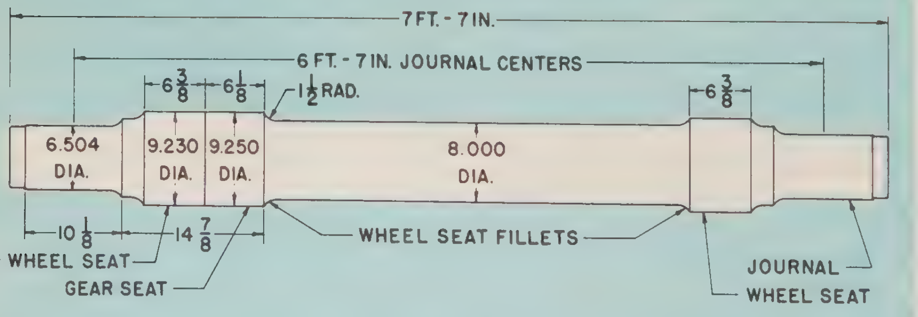


Fig. 1—A typical high-carbon steel railroad axle used by Electro-Motive Division for Diesel locomotives.

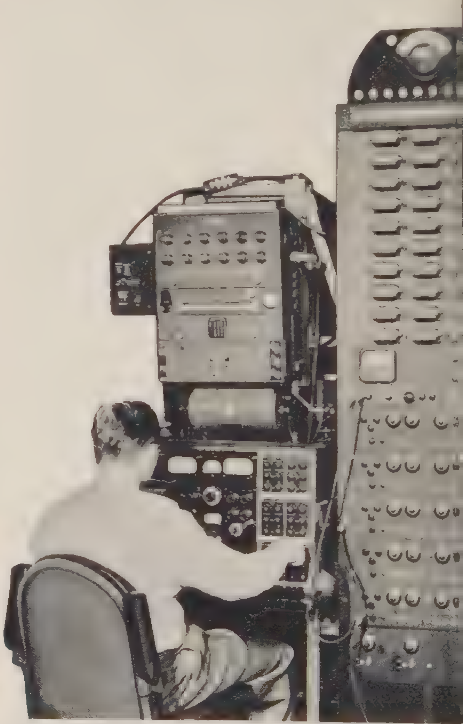


Fig. 2—The Electro-Motive Test Car is well equipped with instruments for many purposes. This picture shows, at the upper left, a 12-channel oscillograph. Underneath is a 12-channel amplifier being adjusted by the test engineer. The tall cabinet contains a 10-channel selective counter. Each of the five-alike trays contains two channels. Above the five is a tray containing the necessary instruments for control and calibration and above this tray are mounted the counters.

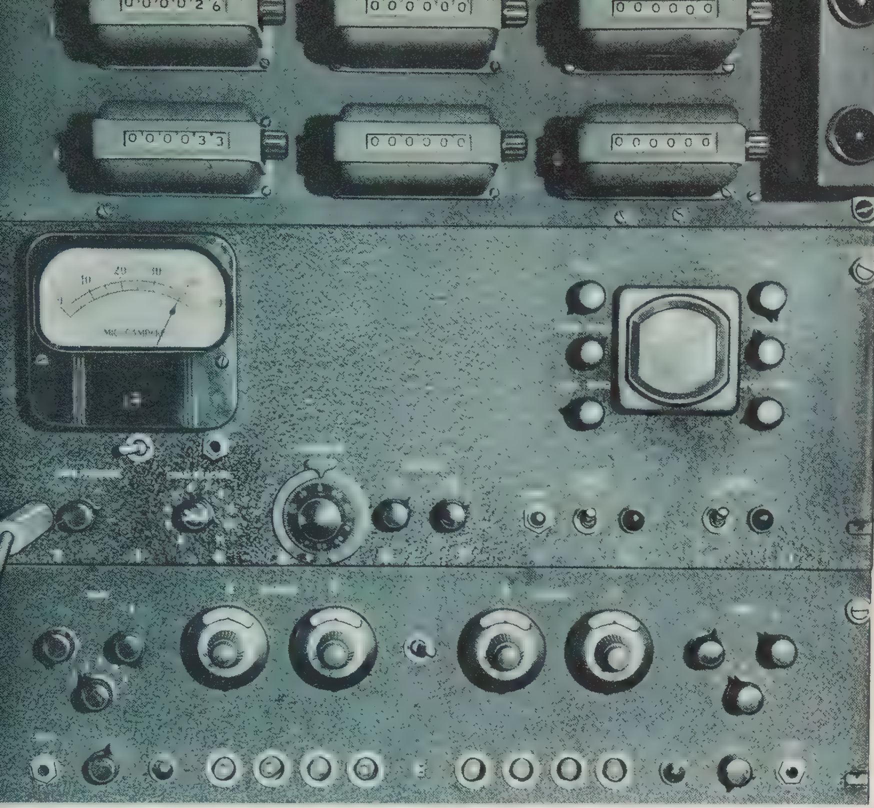


Fig. 3—A close-up of the counter control panel with the counters above and one of the five two-channel panels below. Note the two sets of four small neon lights at the bottom marked "E." One of these light bulbs will light up every time the input reaches one of four preset input levels, but a counter is only provided for each of the upper three levels. Therefore, there are only three counters in each row above—one row for each channel.

Fig. 4—The first indication of failure in a railroad axle usually appears as fine cracks, half an inch to an inch inside the wheel seat. These cracks, shown in the above close-up, may only be 1/16 in. or less deep and they develop very slowly into dangerous proportions.

about the loads an axle is actually exposed to in service.

To determine whether a part of a structure or a machine is strong enough, the engineer must know the type of loading to which it will be exposed. If he does not, he has no other choice than to make a guess and see if it fails. It is too expensive to learn about weaknesses in axles from failures and it is also too expensive to make them so heavy that they are bound to be strong enough in spite of the designers' ignorance about the loads. That is why Electro-Motive Division decided to study axles under service conditions.

Strain Gages

The first requirement for a study of this type is a means for measuring deflection or strain in an axle in service. This is a serious requirement because the axles are exposed to such severe shocks in service that no instrument in use in yesteryears could possibly stand up and give satisfactory readings. The bonded, resistance-type strain gage meets this requirement, but to obtain strain readings with this gage it must be electrically connected to an amplifier and a recording oscillograph and this gets involved because the axle is rotating in a rather inaccessible location underneath a locomotive moving down a railroad track perhaps at 100 mph.

To study axles under service conditions it is necessary to study axles in high speed passenger service and in slow freight service, on curves and on tangent track, on good track and on bad. The axles might be damaged under conditions which occur only occasionally. To make sure that no such condition is overlooked, the behavior of the axles must be observed over long periods.

Test Car

Electro-Motive engineers were in an ideal position to undertake a program of testing axles in the field because of the availability of a railroad Test Car which the Division had built previously. This Car is continuously traveling around the country testing Diesel locomotives under all kinds of conditions and in all types of service. The Car had previously been used for investigating service stresses in locomotive bodies and trucks and was, therefore, already equipped with amplifiers and a 12-channel oscillograph (Fig. 2). The oscillograph is a fine instrument for recording the exact fluctuation of strains over a short period of time whenever the test engineer expects something worth recording to occur, but it is not practical for use over long periods, hours or maybe days, because of the amount of paper which would result. It is necessary to run the recording paper at a speed of several inches per second in order to obtain a clear record.

Selective Counter

Fortunately, the railroad Test Car was also equipped with another instrument which was especially designed and built by EMD as a supplement to the oscillograph. This instrument, the selective counter, does not draw a curve of the fluctuating strain as the oscillograph does, but it counts how many times the strain reaches certain preset levels and it will, therefore, after a trip is completed, show the number of times each of the strain levels has been reached or exceeded. With this information the engineer can draw a probability curve. The instrument can be set to count three different stress levels for ten different gages or it can be set to count six different strain levels for five different

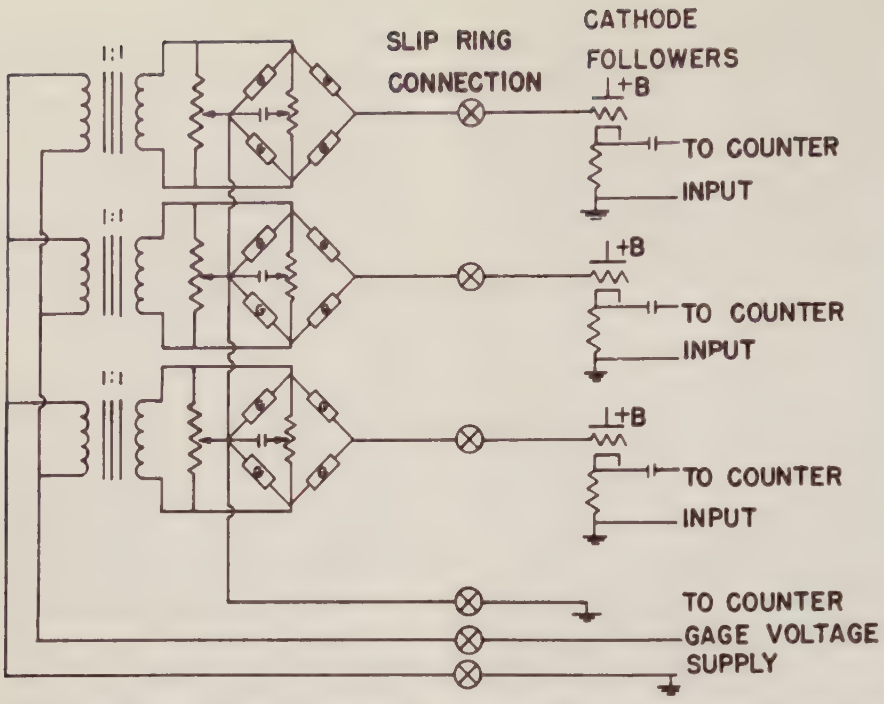


Fig. 5—Wiring diagram for the axle strain gage pickup (only three of the six channels are shown). The transformers and the balancing devices shown with the strain gages *G* to the left of the slip rings were all mounted in a box fastened to and rotating with the axle.

gages. Every time a counter is tripped a little neon light on the front panel lights up and thereby makes the counter a very useful monitor (Fig. 3). Being in constant operation it indicates immediately when high stresses occur and thereby tells the operator under what conditions further investigation with the oscillograph is called for.

Placement of Strain Gages

From axles broken in the past and from available laboratory test reports, it was evident that the weakest point of a normal axle was in the wheel seat, half an inch to an inch inside the inside face of the wheel pressed on the axle (Fig. 4). Another weak point was in the fillet between the axle body and the wheel seat. It was, of course, impossible to place strain gages on the wheel seat underneath the pressed-on wheel and it was difficult to obtain accurate readings with a number of gages located on the fillets because the gages would have to be located accurately due to the high strain gradient at this point. However, this was not necessary either. The nominal moment and stress for any specific load can easily be determined for any part of the axle and the relation between the nominal stress and the actual stress in the wheel seat was known from laboratory tests. Assuming that the stress in the wheel seat is proportional to the moment, it is possible to determine the stress in the wheel seat, or for that matter any place in the axle under service conditions, if the ratio of nominal stress to actual service stress can be measured at any location.

It, therefore, remained only to locate

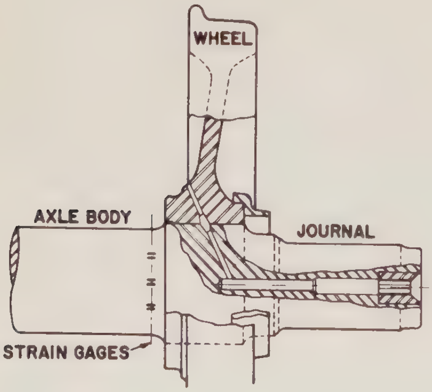


Fig. 6—Cross section of wheel and axle showing the way these parts were drilled to provide a means for bringing the necessary wires from the strain gages and the balancing units out to the end of the axle when they were fastened to slip rings (not shown).

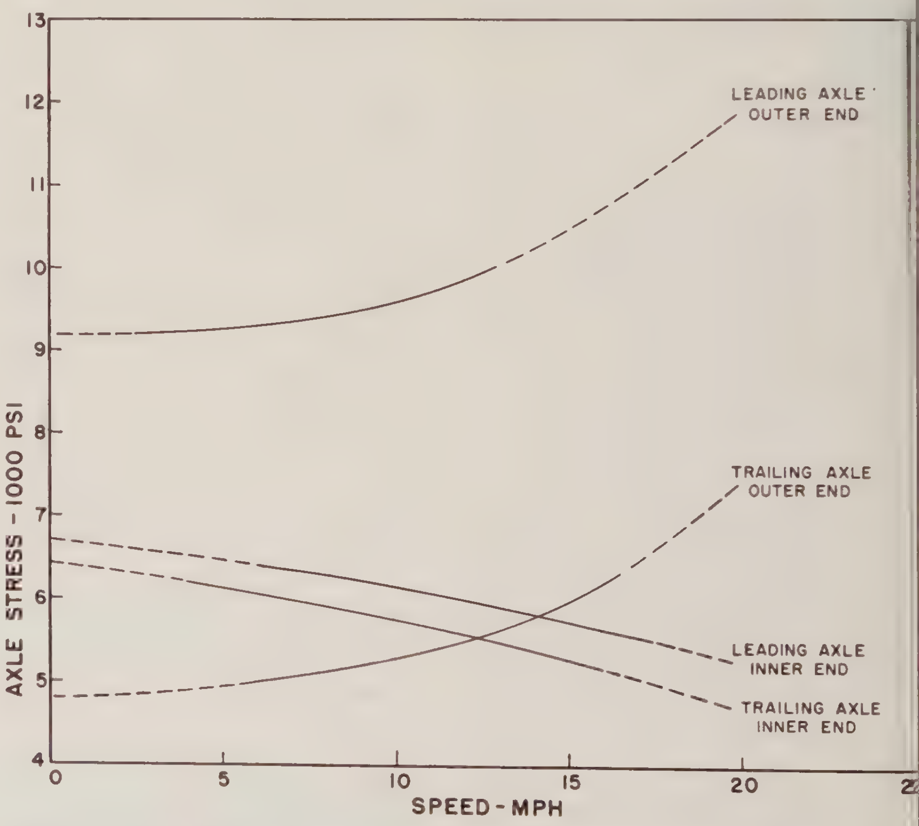


Fig. 7—A truck under a railroad car can roll only in a straight line because the axles are always parallel. It is, therefore, necessary to apply to the outer leading wheel a lateral force large enough to cause the two leading wheels to slide sideways at the same time they roll forward. It is this force which causes the difference between the axle stress in the leading and trailing axles and, since the difference of the ordinates to the two curves is constant, the force does not vary with speed.

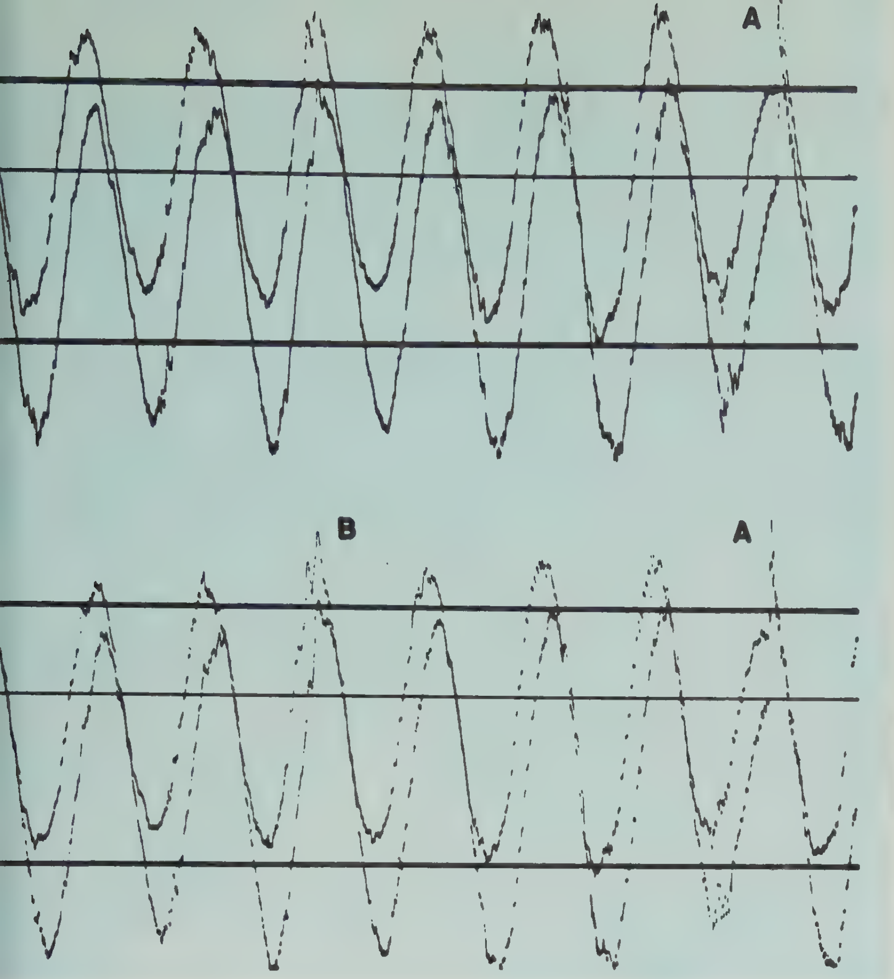


Fig. 8—Two oscillograph records of the axle stress obtained over the same piece of track on two trips. Only two traces and part of a third is shown. Actually, there was one trace for each of the six bridges. The peaks marked *A* and *B* were caused by the two rail joints shown in Fig. 9. (There are four cycles between the two peaks.) A 40-in. wheel is about 10 ft in circumference when new, decreasing with wear. The distance between rail joints is 39 ft.

some strain gages on an axle of the Test Car, bring the leads up to the instrument, and start to collect data.

After some exploratory hookups of gages located at different points of the axle, the investigation was narrowed down to measuring the strain in only one cross section of the axle. For this purpose, six strain gage bridges were mounted on the body of the axle next to the wheel fillet. Each bridge consisted of two gages mounted side by side and two gages 180° away, also side by side. This arrangement resulted in a bridge with four active gages picking up bending strains and eliminating strains due to direct loads and temperatures. As the six bridges were all alike and mounted with 30° spacing between each bridge, a peak stress every 30° of a revolution was obtained and provided, therefore, reasonably good safety against missing important information.

Electrical Connection

The bridge leads were brought to a box mounted on the axle and containing an isolation transformer and a resistance and capacitance balancing unit (Fig. 5). From this balancing box the leads were brought through holes bored through the wheel hub into the center of the axle, out through the center to the journal box (Fig. 6), around a 90° bend, and up to a carbon-silver-on-bronze slip ring mounted on a flexible support on the side of the truck. From the slip rings the leads were brought up in the car where the signal was picked up by a cathode follower before it was fed into the counters and the oscillograph. This wiring resulted in several advantages. The lead wires and slip rings required were reduced to two power leads, one ground lead, and one signal lead per bridge—or a total of nine leads for six bridges. If any noise was generated in the slip rings it only resulted in a reduction in output and, therefore, was harmless. The only exception was the ground lead. A break in this lead could supposedly have resulted in an increase in the signal amplitude and could have been counted as a high stress. For extra safety, two slip rings in parallel actually were used for this lead.

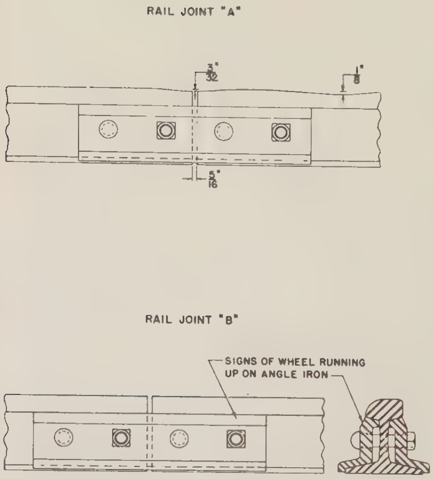


Fig. 9—Two adjacent rail joints demonstrated two ways rails can get worn at the joint. Both caused high stress in the passing equipment (Fig. 8.)

This setup worked very satisfactorily except for a peculiar drift which was troublesome, especially in the beginning. This drift took place almost regularly every time the locomotive to which the Test Car was coupled started or stopped.

Once in motion, the drift was slight. Unfortunately, the bridges could only be balanced when standing still and a man was under the car, but experience soon taught the test engineer how to live with it. Later investigations showed that the dielectric constant for the particular plastic insulation on the leads was very sensitive to temperature changes. Whenever the car was standing still it was exposed to the ambient temperature, but as soon as it started to move, it moved into the warmer air exhausting from the locomotive's traction motors.

Controlled Test

During the early stages of investigating axle stresses in the field, the results obtained proved very confusing.

To clear up two special problems, it was decided to make some tests under controlled conditions on the track located within Electro-Motive Division property at LaGrange, Illinois.

The first problem to be investigated

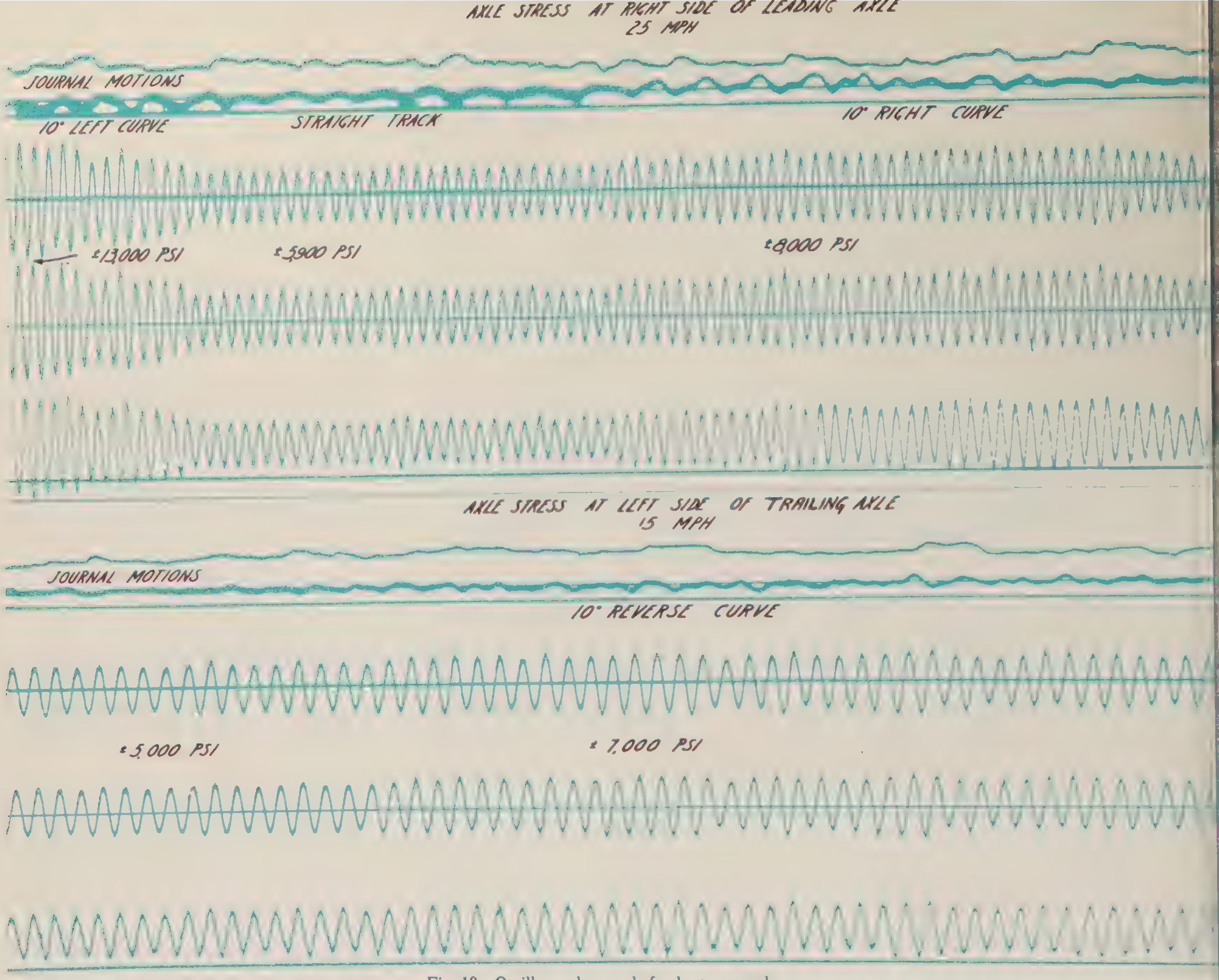


Fig. 10—Oscillograph record of axle stresses when going through a reverse curve on smooth track.

under controlled conditions was the relationship between axle stress and the speed of travel.

It is generally assumed that stresses due to shocks increase in some relation to speed. To check this, the Test Car with its test axle and equipment was repeatedly pulled across a crossover at three different speeds. Unfortunately, it was not possible to exceed 30 mph but, up to this speed, the stresses attributed to shock were proportional to the speed.

The second problem was to establish the stress to which an axle was subjected while negotiating a curve.

Many theories have been advanced regarding the forces needed to make a railroad truck go around a curve. To obtain a picture of what was happening, the Test Car was pulled around a 21° super-elevated curve at various speeds and directions with the test axle in both leading and trailing position. The result of this test is plotted in Fig. 7. It is seen that the axle stress in the outer end of the leading axle is almost a constant amount larger than in the outer end of the trailing axle, which indicates that a force is acting on the outer leading wheel which is independent of speed.

Field Test

Inasmuch as it was unknown under what conditions of service the axle would be most severely stressed, the field test program was based upon subjecting the axle to as many varying operating conditions as possible. This was easily done. While the Test Car traveled around the country running many types of tests on Diesel locomotives in freight and passenger service, the counter was continuously kept in operation counting and indicating high pulses from the axle. The counter levels were adjusted so that normal strains occurring once per axle revolution did not indicate at all, but whenever a high stress occurred it would be counted, the location and condition noted, and, when possible, an oscillograph record taken on a repeat run.

An example of an unknown service condition which subjected the axle to high stress occurred while the axle was tested in freight service. While the Test Car was being run over a track in Michigan it was noted that some unusually high stresses occurred on a certain curve. Oscillograph records were taken on reruns and indicated two peculiar high peaks occurring at the same place on the track each time (Fig. 8). A later inspection of the track revealed the two conditions, shown in Fig. 9, following each other. The splice bar on rail joint *B* was quite worn because the rail was so badly worn that the flange of the wheel rode the splice bar rather than the wheel

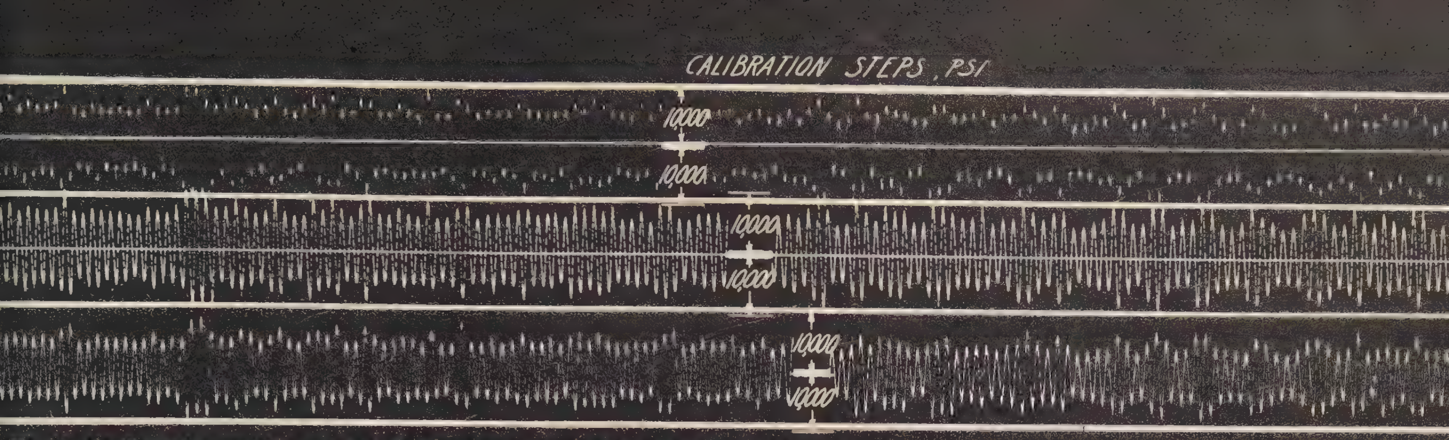


Fig. 11—Oscillograph record from a straight but poor track, causing the car body to roll from one side to the other with accompanying fluctuation in the axle stress. The oscillograph record was taken while the Test Car was traveling at a speed of 15 mph.

tread riding the rail.

As expected, the stress in the axle was always closely related to the condition of the track. Fig. 10 shows a typical oscillograph record of the axle strain taken while the car was going over a reverse curve on a good track. Fig. 11, in contrast, shows an oscillograph record taken while the car was going down a siding with poor track. The car was rolling quite a bit due to the unevenness of the rail so often found on railroad sidings.

Test Results

The axle test program resulted in the collection of many oscillograph records relating to the stress to which railroad axles are subjected under all types of service conditions. Fig. 12 shows an axle stress summary for a leading axle based upon 60 stress records which were selected for variety, reliability, and general interest. Two maximum stresses were plotted for each stress record—the shock stress and the steady state stress. By *steady state stress* is meant stress which stayed above normal for one or more axle revolutions.

The test results in Fig. 12 are arranged according to the speed at which they were obtained. It is obvious from an inspection of the plot that, contrary to the results obtained during the controlled shock-speed test, there is no relation here between the speed and the stress. This can, of course, be explained by the fact that the speed of the train is restricted by the railroad to suit the track or, stated in another way, that the railroad maintains their track according to the speed at which they wish the trains to travel.

The frequency with which the stress reached the preset stress levels on various runs was obtained from the selective counter. Studying these results it soon became clear that there was a relationship between the frequency of high stresses and the geographical location of the railroad where the records were obtained.

Railroads in mountainous territory necessitating many sharp curves were distinctly different from railroads on the plains.

The count obtained on two of the most extreme railroads in this respect were plotted (Fig. 13). Most other railroads will be found to fall somewhere in between. It should be noted that there is a remarkable difference in the number of times the intermediate stresses are reached on a track with many curves and on a track with few curves, but there is little difference between the frequency with which the peak stress is reached on the two types of track.

Analysis of Test Data

With the collected field data it only remains to correlate the data with various laboratory tests reported by others. This is made difficult because the field test gives stresses varying all the way from less than normal to the peak, whereas the laboratory tests are nearly all based upon a constant load for each axle tested. The fatigue limit is always an uncertain factor regardless of the type of loading because it is influenced by so many factors, such as surface finish and built-in stress concentration. The Association of American Railroads has come to the following fatigue limit based upon many tests:

- Chrysler Super Finish	31,000 psi
- Smooth or Rough Turned	28,000 psi.

These values are low compared with many other published results and should be safe for axle design purposes.

The above fatigue limits are, however, established by test with repeated constant load whereas the actual loads to which the axles are exposed in service are very irregular. From results of investigation published in this area (T. J. Dolan, F. E. Richart, Jr., and C. E. Works, "The Influence of Fluctuations in Stress Amplitude on the Fatigue of Metals," *Proceedings—Committee Reports, Technical Papers*, American Society for Testing Materials, Vol. 49, (1949), pp. 646-82), the following conclusions may be reached for axle steel:

- When the majority of stress cycles (minor stresses) lie below the endurance limit with the major stresses above, the stress level of the endurance limit appears to be about 10 per cent higher than would be predicted by constant loading.
- With loading conditions as above, the cycle life N will be less than predicted (from normal S-N curves obtained from constant loading) by completely ignoring the minor stresses, but greater than predicted by considering the minor stress cycles as equally damaging, that is, considering N to be the sum of the major cycles plus the minor cycles.
- With loading conditions such that the minor stresses are slightly above the endurance limit and the major stresses considerably above, the cycle life is the same as though the major stresses were equal to the minor stresses, that is, occasional high stresses are not damaging (unless they exceed the yield point).

AXLE STRESS SUMMARY

(LEADING AXLE)

B=BRAKING R=ROLLING	TRAIN SPEED M.P.H.	SUPER ELEVATION INCHES	DEGREE OF CURVE	TRACK C=CURVE S=STRAIGHT	AXLE END O=OUTER I=INNER
R	4	0	0	S	—
R	10	0	0	S	—
R	10	—	—	C	I
R	10	—	—	C	O
B	11	2	8.5	C	O
B	12	2	8.9	C	O
R	12	0	0	S	—
R	12	—	5	C	O
R	12	0	0	S	—
R	12	—	5	C	I
R	15	0	0	S	—
R	15	—	—	C	O
R	16	3.5	5	C	O
R	20	—	10	C	O
R	20	—	—	C	I
R	20	0	0	S	—
R	20	0	0	S	—
R	20	—	—	C	O
R	22	—	—	C	O
R	22	—	—	C	I
R	28	—	—	C	O
R	28	—	—	C	I
R	30	3.5	3	C	O
R	32	3.5	5	C	O
R	32	3.5	7	C	O
R	33	2	4.5	C	O
R	34	3.5	4	C	O
R	34	3.5	4	C	O
R	34	—	—	C	O
R	34	—	—	C	I
R	35	3.5	3	C	O
R	35	3.5	3	C	I
R	35	3.5	3.3	C	O
R	35	3.5	5	C	I
R	37	0	0	S	—
R	37	0	0	S	—
R	38	0	0	S	—
R	38	0	0	S	—
R	39	3.5	3	C	O
B	40	0	0	S	—
R	42	—	—	C	I
R	42	—	—	C	O
R	42	—	—	C	I
R	42	—	—	C	O
R	43	—	6	C	O
R	44	—	—	C	I
R	44	—	—	C	O
R	45	—	—	C	I
R	45	—	—	C	O
R	50	—	—	C	O
R	60	—	—	C	I
R	75	—	—	C	I
R	90	—	—	C	O
R	90	—	—	C	I
R	90	—	—	C	O
R	90	—	—	S	—
R	95	—	—	C	I
R	95	—	—	C	O
R	96	—	—	S	—
R	100	—	—	S	—

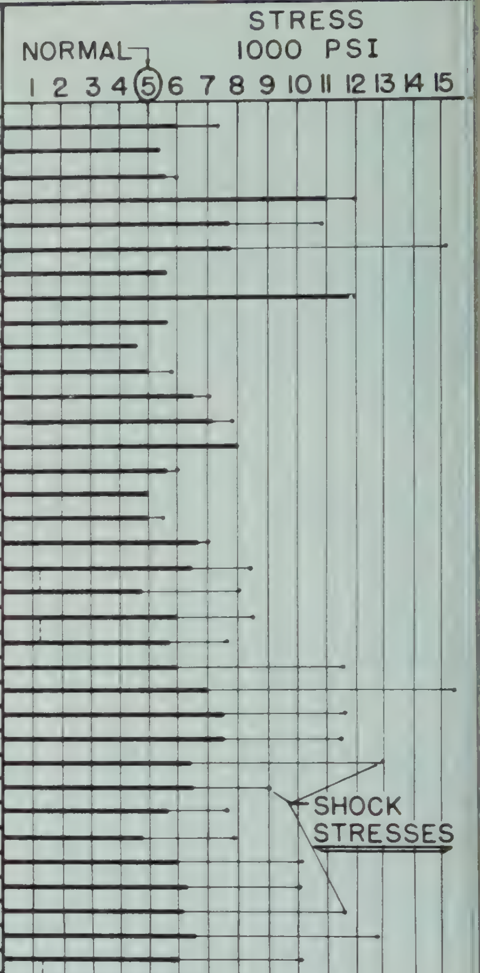


Fig. 12—A graphic presentation of the peak stresses found on 60 oscillograph records and arranged according to the speed. The heavy lines indicate stresses due to peak loads which lasted for a full turn of the axle or more, such as the stress shown in Fig. 10. The dots at the end of the light lines indicate peak stresses of short duration such as the shock stresses shown at *A* and *B* in Fig. 8. Note that there are as many high stresses at low speed as there are at high speed.

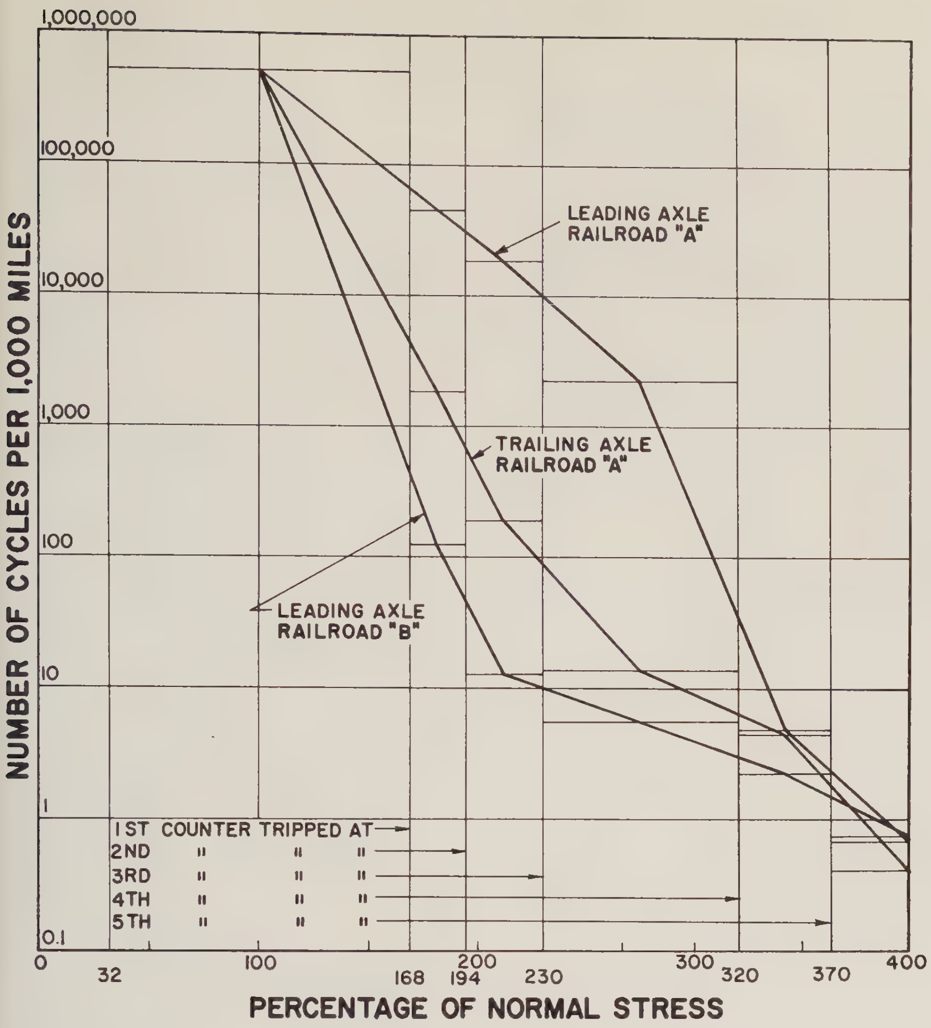


Fig. 13—The curves show how many times a certain stress (given in per cent of normal stress) can be expected to be reached during a 1,000-mile run over two extremely different types of track. Railroad A is a mountain road with many sharp curves. Railroad B is on the plains with few and only light curves.

This would indicate that the fatigue limits as established by the A.A.R. can be increased 10 per cent for axles and that the very few extreme stresses can be neglected.

The wheel seat of EMD's axles are smooth turned and must, therefore, be assumed to have a fatigue limit of 28,000 + 10 per cent = 31,000 psi in service. The body and fillet of the axle are ground and it must, therefore, be assumed that the fatigue limit in service is approximately 31,000 + 10 per cent = 34,000 psi for the body.

Electro-Motive Division has found experimentally that the stress concentration in the wheel seat fillet is about 1.5.

The A.A.R. has demonstrated that a nominal stress above 9,000 psi in the wheel seat will develop fatigue cracks. This indicates an equivalent stress concentration factor of 28,000/9,000 = 3.11 in the wheel seat.

If the axle is installed in a four-axle Diesel locomotive weighing 248,000 lb on the rail, then the bending moment in the axle causes a nominal bending stress of 3,780 psi in the wheel seat and 6,080 psi in the wheel seat fillet. Using the strain concentration factors of, respectively, 3.11 and 1.5, the actual normal stress in the axle is 3,780 × 3.11 = 11,800 psi in the wheel seat and 6,080 × 1.5 = 9,120 psi in the wheel seat fillet.

If the nominal stress in the wheel seat is 11,800 psi and the fatigue limit 31,000 psi, then the fatigue limit in the wheel seat is 262 per cent of the nominal stress. The same way, if the nominal stress in the wheel seat fillet is 9,120 psi and the fatigue limit 34,000 psi, then the fatigue limit of the wheel seat fillet is 372 per cent of the nominal stress. Referring to Fig. 13, it is seen that for a leading axle in hard service (Railroad A) the actual stress will be 262 per cent of the nominal stress 4,400 times, and 372 per cent of the nominal stress two times—for each 1,000 miles the locomotive travels.

Assume that a Diesel locomotive travels 300,000 miles per year and that the axle is in a leading position half the time. The axle makes approximately 500 revolutions per mile. This means that the axle will go through 75,000,000 revolutions per year in a leading position and, therefore, that the wheel seat will be stressed up to the fatigue limit 300 × 0.5 × 4,400 = 660,000 times per year and the wheel seat fillet 300 × 0.5 × 2 = 300 times per year.

It requires several million cycles to produce a crack with a stress close to the fatigue limit but, unfortunately, there is a large spread in the number of cycles determined experimentally.

It must, therefore, be concluded that there is a possibility of cracks developing in the wheel seat of an axle after several years service. The A.A.R. has, however, also shown that there is a large margin in the number of cycles between the initial crack and the ultimate failure. The axles are checked for cracks every time the wheels are removed which happens every one or two years when the wheels are worn out and it is, therefore, unlikely that a complete failure in the wheel seat will ever occur.

The stress in the wheel seat fillet will be stressed up to the fatigue limit only 300 times per year and it must, therefore, be concluded that the axle will never fail in the fillet unless first damaged by other factors, such as hot bearings.

Conclusion

It must be admitted that, in spite of the tremendous amount of data collected over the span of about two years, one still may not predict with certainty what stress levels will be reached, or more important, what the fatigue life will be for any particular axle.

However, this is not to say that such tests are without value. On the contrary, any information concerning service conditions is always in great demand by the engineers who must design the equipment, for the more information available, the lesser the chance of either overdesigning or underdesigning.

How Industrial Suppliers and Design Engineers Work Together

By WALTER T. BURWELL
Chevrolet Motor
Division

The industrial supplier and the manufacturer—mutual participation for shared benefits

It is inconceivable that any one manufacturing organization—no matter how large—could have an engineering staff and facilities extensive enough to cover all the detailed engineering aspects of a complex mechanism, such as an automobile. Consequently, all industrial manufacturers depend, to a considerable degree, upon parts and materials suppliers for engineering information. As a result, interdependence among industries assumes a greater role every day and, today, the cooperation between the industrial supplier and the manufacturer of the finished product is more important than ever. The manufacturer's design engineer is keenly aware of this situation and is eager to learn of new developments in any field which might improve his product. In turn, the industrial supplier desires to strengthen this relationship between himself and his customer's design engineers. The result—a freer exchange of ideas leading to a better product, more efficiently produced at a lower cost.

The industrial supplier is, in a very real sense, the business partner of the manufacturer who uses his product. His counsel is sought by the manufacturer's design engineers since both have the same ultimate goal—to provide a better end product at a lower cost. Perhaps the best way to illustrate this fact is first to describe a typical industrial organization and then to outline the relationships between the manufacturer's design engineers and the industrial suppliers.

For example, in the Chevrolet Motor Division, only a portion of the total engineering design is handled in the Central Office Engineering Department in Detroit. A great number of design modifications results from the work of production engineering staffs located at various Chevrolet plants in scattered locations. These design changes—proposed to permit more efficient use of equipment or to set up some new method of manufacture—often are no less important than the original design work done in the central Engineering Department.

An objective of the automotive design engineer is to establish a design which can be manufactured economically in large volume. This requires the closest cooperation between the Central Office design engineer and the engineers stationed at the Division's many manufacturing and assembly plants. One of the most important duties of the resident engineers—those assigned to the manufacturing plants—is to keep Central Office engineers informed of manufacturing and production problems. Parallel with the duties of the resident engineers in manufacturing plants, those assigned to engineering control in the Division's assembly plants report assembly problems to the Central Office design engineers.

Combined information from these two sources permits design changes to be made to benefit current production and also serves as a guide in establishing new designs for future production.

When a new design is offered for production, the production engineer and the resident engineer at the manufacturing plant concerned are advised of the proposed design. Together they establish dimensional and material specifications as well as the physical characteristics required for the proper functioning of the part. In many cases, this cooperation avoids costly and time-consuming changes after release for production.

By the time the part is actually released for production, the resident engineer is well informed of what is expected of the part and can advise the production department of critical points to be considered in tooling for the job. At this point, the resident engineer works very closely with management at his plant while decisions are made on what types of equipment and which methods of processing will be used.

Often, the production department requests minor changes on the present equipment to aid them in processing the part. These changes are proposed to assure better quality, to permit the use of less expensive tooling, to allow the use of less critical material, to reduce overall cost, or to permit easier handling and shipping.

From this example, it is apparent that there is no sharp division between design engineering and production engineering responsibilities as both work together toward the same objectives. In fact, the production engineer, in many instances, is a design engineer, inasmuch as he can suggest design changes.

This example of the breakdown of engineering responsibilities at Chevrolet Motor is indicative of the setup in almost every large industrial organization, particularly those in which the manufacturing is decentralized.

The Industrial Supplier and Design Modifications

In large volume production, it is not unusual for several outside sources of manufacture to be used, especially for small parts, such as piston rings, oil seals, and gaskets. (General Motors, as a whole, uses over 21,000 suppliers who provide materials and services.) Information on the parts drawings released to suppliers is limited to those specifications desired in the finished piece. The processing details are left to the supplier, in order to permit the maximum amount of latitude in processing.

As it is unlikely that these various sources will have identical equipment available, each source establishes somewhat different methods of processing the part or parts it supplies. Also, it is very probable that one particular design will not lend itself to the most economical manufacture at each and every source. Consequently, optional constructions often are shown on the original drawings made by the design engineers.

It is common practice—when more than one supplier is producing a given part or assembly—for optional constructions to be used (Fig. 1). The only requirements are that these parts must be completely interchangeable and must have equivalent performance. There are examples of such optional constructions in Chevrolet Motor Division itself. Specifically, the Flint, Michigan, and Tonawanda, New York, plants produce parts for the same engine but have different manufacturing facilities. Because the processes vary, a camshaft made in Flint may not be identical with one made in Tonawanda—but camshafts from the two cities are fully interchangeable in the engine.

Other examples of optional designs and optional materials are found in such items as bearings, brake linings, and gaskets. Whenever options are used, the part is tested to assure interchangeability, equivalent performance requirements, and durability.

It is impossible for the design engineer who makes the original parts drawing to anticipate fully the manufacturing source's problems; in many cases, he does not even know who the supplier is to be.

To remedy this deficiency, the industrial supplier cooperates with the plant resident engineer in much the same way as the Central Office designer and the resident engineer function together. The supplier analyzes the design submitted to him for manufacture and suggests any changes to the resident engineer which he feels will promote a better product at a lower cost—the mutual aim of both the manufacturer and the industrial supplier. Occasionally, he prepares alternate drawings and submits them to the Central Office design engineer.

But, before attempting to suggest design changes, the supplier studies the proposed product and its functions. Will his design modification affect the product's interchangeability, lower its performance, weaken its durability, or raise its cost? If the answer is in the negative, the supplier and the original design engineer go over the proposed design modifications together. In most cases, the result is a design change which satisfies the production source without sacrificing any of the original characteristics set up by the design engineer.

Design analysis and processing studies on the part of the supplier continue even after the part is in actual production, since experience gained in producing the item may dictate the advisability of minor design revisions either to improve the product still more or lower the cost without reducing the quality.

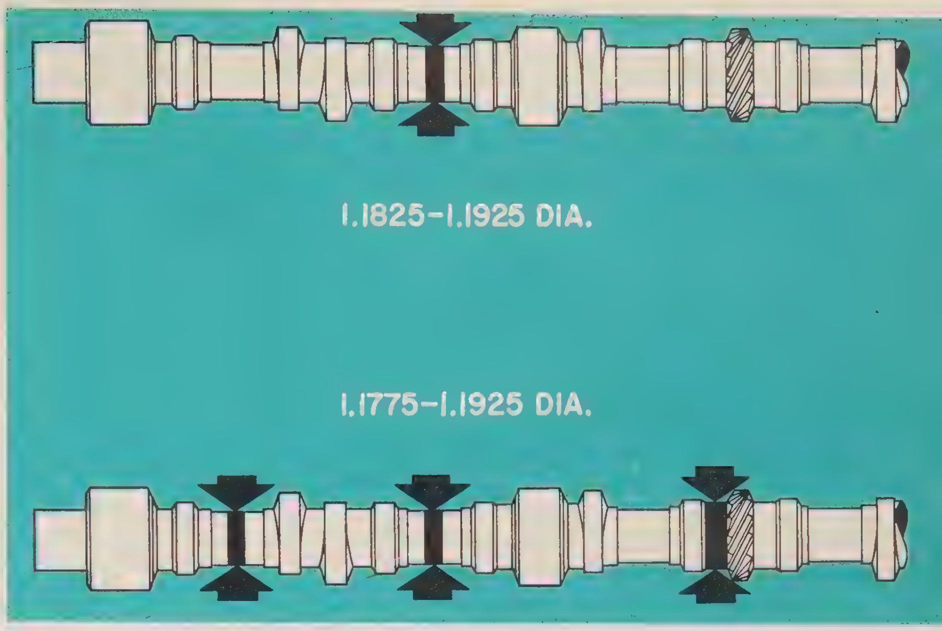


Fig. 1—Often, more than one supplier furnishes the manufacturer with the same product. As it is improbable that any two of these sources have identical processing equipment available, design changes are sometimes made to facilitate manufacture. The two camshafts shown have different specifications for the surfaces (arrows) used only to position the shaft when machining the cam, bearing, and gear surfaces. In each case where such options must be used due to differences in processing equipment, the part is carefully inspected to assure complete interchangeability.

The Supplier Offers His Own Product

Sometimes, a particular supplier develops a product of his own design which he feels the manufacturer may be able to use. He has already familiarized himself with the manufacturer's end product or products and he may visualize one or more possible applications for his new product. He submits his new design to chemical and physical tests and compiles all the pertinent data. He smooths out as many of the rough spots in his design as he can.

His next step is to contact the manufacturer's design engineer. These contacts are on an engineering basis rather than a sales basis. (As a rule, the contact is made by the supplier's sales engineer or by the supplier's engineer and sales representative together. This way, engineering information can be exchanged readily and a minimum of time is consumed. Recently, the trend has been toward technically trained sales representatives or at least salesmen with a working knowledge of engineering fundamentals.)

The sales engineer (or sales representative) comes to the meeting prepared with data on his new product's

- Design criteria
- Physical characteristics, such as resistance to corrosion, resistance to oil and gasoline, and effect of high and low temperature operation
- Processing limitations
- Handling problems
- Typical applications
- Servicing procedure
- Patent restrictions.

(While all of these data may not be applicable to some products, the supplier's information is as complete as possible in every case. The reason is that, even if the supplier suggests a particular application for his product, the disclosure of these data may prompt the design engineer to consider the new product for another application he has in mind—one that the supplier did not recognize.)

One effective method of presenting data that is sometimes used by the supplier is a chart showing both the advantages and the disadvantages of a proposed product. Table I is an example of one of these charts. (The terms in this table are of a general nature for illustrative purposes; in an actual presentation the features are as specific as possible.)

The alert supplier keeps in mind that the design engineer is trained to deal with facts and to face realities, but that

COMPARISON OF FEATURES OF PROPOSED NEW PRODUCT

ADVANTAGES	DISADVANTAGES
Improved Durability	Slight Weight Increases
Lower Handling Costs	
Less Scrap Loss	Limited Function at —40° F
Lower Maintenance Cost	Requires Modification of an Adjacent Part
Lower Cost	
More Uniform Quality	

Table I—One of the methods sometimes utilized by the industrial supplier's sales engineer for presenting the strong and weak points of his new product is illustrated in this table. The industrial supplier endeavors to turn over to the manufacturer's design engineer a product as completely developed as possible—requiring, on the part of the manufacturer, only tests to verify data on the product's performance and durability characteristics as listed in a chart of this type. (The terms of this chart are of a general nature for illustrative purposes; in actual presentation, the features are as specific as possible.)

he is always aggressive and seeking new and different ways to do things. Because of this, the design engineer is curious and somewhat skeptical—or at least to others he sometimes seems skeptical. He must assemble data in an unbiased manner and study them before arriving at a conclusion. All of these characteristics are summed up in what is called "a scientific attitude of mind." Consequently, the supplier knows that the most effective method for gaining the design engineer's confidence is to submit factual data regarding his new product and its performance. So, the supplier is quick to label a fact as a fact and to have factual data on hand to back up an opinion. In addition, the supplier does not hesitate to disclose the limitations of his product for any given application—in advance of any tests run by the design engineer. This procedure avoids delays and expense that occur later when the limitations are discovered by the manufacturer and the honesty of such a presentation leaves a much more favorable impression with the design engineer.

Summary

The industrial supplier and the manufacturer work together in the same sense as business partners cooperate. The industrial supplier is encouraged to offer practicable design changes and to submit new products. The manufacturer's design engineers are kept up-to-date on engineering information and the supplier's processing problems. And this cooperation, in turn, results in a more efficient finished product at a lower cost.

The Importance of Liaison Between Engineering and Patent Departments

It is generally the policy of manufacturing corporations to provide for close liaison between their engineering groups and their patent attorneys to insure that all new or materially improved devices be considered from a patent standpoint, preferably before these devices go into production. Often, however, this close liaison is not continued until the time when a patent is granted but ceases after the patent investigation has been made, with the result that the fullest patent protection may not be obtained.

If it appears that a new or improved device is patentably novel, a patent application may be filed, including drawings illustrating the device, a specification describing it, and one or more claims pointing out the subject matter which is regarded as the invention. Generally, this application remains pending in the U. S. Patent Office for at least three years, and often for four or more years, before the patent issues. In the course of prosecution, the pending application may be amended. Once the patent has issued, its scope of protection depends primarily upon the terminology of the patent claims.

During the time that the patent application is pending in the Patent Office, it often happens that the device forming the subject of the patent application is changed. Under such circumstances, when the patent issues, its claims may not cover the changed device at all or the claims may not cover the changed device in the best possible manner—particularly if the prosecution of the patent application was carried on without knowledge of or regard for engineering changes in the device.

Application May Be Amended

If close liaison is maintained between the engineering activity and the patent attorney during the pendency of the application, it is often possible to amend the patent application to keep it abreast of the engineering developments and changes relating to the device so that the patent, when issued, will cover the device to give the fullest possible protection even though the device may be changed in some respects from its original form.

The Rules of Practice of the Patent Office governing the prosecution of patent applications provide that the specification or description "describe completely a specific embodiment of the process, machine, manufacture, composition of matter or improvement invented . . ." In addition, the specification must explain the mode of operation, and the best mode contemplated by the inventor for carrying out the invention must be set forth. Neither the patent statutes nor The Rules of Practice of the Patent Office require that all possible modifications be disclosed or described.

In amending a patent application, the insertion of *new matter* not originally disclosed is prohibited. All amendments to the drawings or the specification and all additions thereto must conform to either the drawings or the specification as originally filed and new matter which was not found in the original specification or drawings and which involves a departure from or an addition to the original disclosure cannot be added to the application by amendment. Because of this prohibition against the addition of new descriptive matter, care must be exercised by the patent attorney to avoid the insertion of new matter in revising the specification or the drawings. Generally, if an adequate description was originally filed, it is better to leave the specification and drawings in their original form than to attempt to modify them to include engineering changes in the device. However, the terminology of the claims often may be broadened or limitations which are unnecessary to patentability may be deleted so that the claims will cover not only the device as originally shown and described in the patent application but also the device in its changed form.

Close Liaison May Improve the Patent

For example, assume that chairs and other seating devices of any kind were completely unknown until the invention of a simple three-legged stool. A patent application filed on this device included drawings of the stool, a specification describing its construction and use, and several claims defining the invention. Since neither the inventor nor the patent

By WILLIAM S. PETTIGREW
Patent Section
Central Office Staff

Working together, the engineer and attorney often broaden the scope of the original claim

attorney had knowledge of any other seating devices or any indication of possible improvements, the attorney inadvertently included in the claims certain details and limitations which were quite unnecessary to the patentability of the device, and which would restrict the coverage of the patent insofar as possible improvements in the device were concerned as, for example, limitations to "three legs" and to "a horizontal support." In experimenting with the three-legged stool, it became apparent to the inventor that the addition of a fourth leg would stabilize the stool and make it less subject to tipping and that the addition of a vertical back support would make the stool more comfortable. Amendment of the patent drawings or the specification to illustrate and describe these improvements would be prohibited as being new matter. But if the patent attorney were kept advised of these improvements, he could modify some or all of the claims to eliminate the unnecessary limitation of three legs by defining the legs more broadly as, for example, "a plurality of legs." The claims also could be modified to remove the unnecessary limitation to a horizontal support by defining the seat and back in a single generic term, as for example, "supporting means."

Through close liaison between the inventor and the patent attorney and through skillful prosecution on the part of the attorney, the patent, when issued, would contain claims limited only to support means and a plurality of legs upon which the support means is mounted. Such claims would cover the stool as originally invented and would also cover the improved device having four legs and a vertical back support.

Of course, the above example is an oversimplification of the problems in obtaining proper protection. However, close liaison while the application is pending can, in the same manner, result in fuller protection.

Notes About Inventions and Inventors

Contributed by
Patent Section
Central Office Staff

On this and the following pages are listed some of the patents granted to General Motors during the months of November 1954 to January 1955. The brief patent descriptions are informative only and are not intended to define the coverage which is determined by the claims of each patent.

Patents Granted

- **Jakob A. Adloff, Adam Opel A.G.,** *Russelsheim/Main, American Zone, Germany, for a Gearshift Mechanism for Transmissions, No. 2,693,714, issued November 9.* This invention covers a single link connection between the gearshift control linkage on the mast jacket and the linkage in the transmission housing to transfer the rotary and axial movement of the transmission control linkage on the mast jacket to the selecting and shifting linkage in the transmission.

Mr. Adloff is head of the Development Section of Adam Opel's Engineering Department. In 1927, directly after earning the engineering degree from Rheinische Ingenieurschule, Bingen, he joined Adam Opel as an inspector in the Gear Inspection Section of the Production Department. In 1928, he transferred to the Chassis and Engine Design Section of the Engineering Department. One year later he was made a detailer, promoted to designer (1930), to group leader (1931), to responsible development engineer (1935), and to his present position in 1940.

- **Karl Jochem,** *Adam Opel A. G., Russelsheim/Main, American Zone, Germany, for a Remote Gear Control for Motor Vehicles, No. 2,693,715, issued November 9.* This patent is directed to a transmission control linkage consisting of a control tube mounted externally on the steering column assembly, operated by a shifter handle pivoted to the steering column assembly and the control tube at the upper end, and actuating a similar lever on the steering column assembly and the control tube to transfer the movement of the shifter handle to the controlled lever at the base of the steering column assembly.

Mr. Jochem has been passenger car chassis engineer in the Engineering Department of Adam Opel since 1954. Starting in 1925 as a detailer in the Chassis Section of this Department, he was promoted through positions of designer for chassis components, checker, group leader for truck design, and head of the truck drafting room and assistant truck engineer at Brandenburg and Russelsheim, to head of the entire chassis design for cars and trucks, including the engine. Mr. Jochem received his engineering degree in 1924 from Rheinische Ingenieurschule, Bingen.

- **Clair A. Short, Jr. and Leslie R. Smith,** *Allison Division, Indianapolis, Indiana, for a Control of Aircraft Engines, No. 2,695,014, issued November 23.* This patent covers a manifold pressure regulator for aircraft engines in which selected manifold pressures are automatically maintained and engine speeds controlled automatically in accordance with a plurality of schedules, the regulator having means for selecting the different speed schedules to be maintained with the selected pressure schedule.

Mr. Short is section head of Production J71 Engines in Allison's Turbo-Jet Engineering Section. He holds both the B.S.

*Inventors' names marked with an asterisk in this section have had their biographies published in a previous issue of Volume 2, GENERAL MOTORS ENGINEERING JOURNAL.

degree in mechanical engineering (1934) and the M.S. degree in mechanical engineering (1936) from Georgia Institute of Technology. He joined Allison in 1943 as a detail engineer and was promoted to project engineer in 1944, to senior project engineer in 1947, to engine project engineer in 1951, and to his present position in 1954.

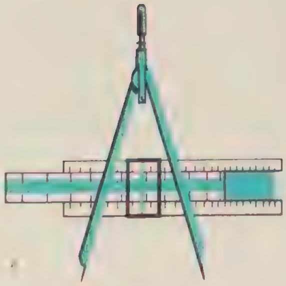
Mr. Smith is chief draftsman in the Turbo-Jet Engineering Section of Allison. In 1942 he joined the Division as a detailer in the Drafting Department and has since been promoted through the positions of layout man (1945), checker (1946), project engineer (1947), and senior project engineer (1951), to his present position, chief draftsman (December 1951). Mr. Smith completed engineering courses from International Correspondence School.

- **John Dolza,** *General Motors Engineering Staff, GM Technical Center, Detroit, Michigan, for a Combined Torque Converter and Engine Air Cooling System, No. 2,696,074, issued December 7.* This patent covers a cooling system operating to provide cooling for an engine and also for the fluid circulated by a torque converter or the like.

Mr. Dolza is engineer-in-charge of Power Development at the GM Central Office Engineering Staff, in which capacity he has served since 1945. Much of the work of this group is related to automotive and military engines, air conditioning, and refrigeration. From 1940 to 1945, he was a consulting engineer at Allison Division, engaged in development of various automatic controls for aircraft and turbo-prop engines. Prior to this time, Mr. Dolza was associated with Buick Motor Division, starting in 1927 as a draftsman and advancing to assistant chief engineer. He attended the Polytechnico Institute, Turin, Italy, from 1920 to 1926, receiving the M.S.E.E. and M.E. degrees.

- **William H. Flowers and Robert S. Plexico,** *Chevrolet Motor Division, Detroit, Michigan, for Transmission Anti-Rattle Gearing, No. 2,696,124, issued December 7.* This patent covers an arrangement whereby a gear splined to a shaft is prevented from rattling when power is not being transmitted by the gear.

Mr. Flowers is chief production engineer in the Chevrolet Motor Engineering Department. Originally employed by the Division in 1919 as a draftsman, he



became chief draftsman in late 1919, engineer in 1920, transferred to the Chevrolet Gear and Axle Plant in 1922, returned to Chevrolet Central Office in 1941, and was promoted to his present position in 1943. His educational background includes business training and mechanical engineering.

Mr. Plexico is assistant chief engineer at Chevrolet. He came to this Division in 1934 as a project engineer in the Engineering Department. He later advanced to division engineer, assistant staff engineer, and staff engineer. His projects have included design work on new trucks as well as clutches and transmissions. Five patents have resulted from this work. He received the B.S. degree in chemistry from Clemson A. & M. College in 1919 and had automotive experience with three other firms before joining Chevrolet. He is a member of S.A.E. and has served on its Truck and Bus Committee.

- **Edward P. Harris,** *Inland Manufacturing Division, Dayton, Ohio, for a Device for Applying End Rings to Hose or the Like, No. 2,696,239, issued December 7.* This invention relates to an apparatus for placing a bead or finish ring around the end of flexible defroster hoses to prevent unraveling of the hoses.

Mr. Harris serves as a project engineer in Inland Manufacturing's Engineering Department, and is concerned mainly with foam door seals and car air conditioning. He joined Inland in 1931 as a student in training. From 1938 to 1942 he was technical supervisor at the Inland Clark Plant and, from 1942 to 1945, a laboratory supervisor at Eastern Aircraft—a part of General Motors during World War II. Mr. Harris received the M.E. degree from Cornell University in 1931. His technical affiliations include membership in the Society of Automotive Engineers and the American Society for Metals.

- **C. W. Lincoln,* Philip B. Zeigler,* and Henry D. Spiekerman,** *Saginaw Steering Gear Division, Saginaw, Michigan, for a Transmission Control, No. 2,696,126, issued December 7.* This invention relates to a linkage mounting and cam arrangement providing an improved linkage connecting the hand lever and the controlled levers of a transmission control mounted on the steering column.

- **C. W. Lincoln,* Philip B. Zeigler,* and Ralph A. Malone,** *Saginaw Steering Gear Division, Saginaw, Michigan, for a Transmission Control, No. 2,696,127, issued December 7.* This invention relates to a linkage mounting and cam arrangement providing an improved linkage connecting the hand lever and the controlled levers of a transmission control mounted on the steering column.

Mr. Malone is a designer in the Product Engineering Department of Saginaw Steering Gear. Employed by this Division in 1942 as a junior tool designer, he was promoted to senior tool designer in 1943 and to his present position in 1948. Mr. Malone's previous work with steering, transmission controls, and signal switches has resulted in five granted patents. Presently, he is concerned with advancements in power steering. Mr. Malone is a member of the Society of Automotive Engineers.

Mr. Spiekerman is no longer with the Division.

- **Dwight M. Overcash and Richard B. Parks,** *Brown-Lipe-Chapin Division, Elyria, Ohio, for a Copper-plating Bath and Process, No. 2,696,467, issued December 7.* This patent relates to an acid copper-plating bath containing thiourea or a derivative of thiourea and lignin sulfonic acid or a salt thereof. The bath produces bright deposits of copper.

Mr. Overcash is plant manager of Brown-Lipe-Chapin's plant in Elyria, Ohio. Originally employed by GM in 1934 as a laboratory assistant at Guide Lamp Division, he transferred in 1936 to Brown-Lipe-Chapin where he became foreman of the Plating Department in the Division's Syracuse, New York, plant. He became a supervisor of process engineering in 1942 and transferred to Elyria in 1947 as general superintendent. He became factory manager in 1951 and, five months later, was promoted to his present position. Mr. Overcash received his B.S. degree in 1932 from Ball State Teachers College and his M.A. in 1939 from Indiana University.

Mr. Parks is a process engineering supervisor for Brown-Lipe-Chapin. He joined the Division in 1948 as a laboratory tester. He subsequently was made Plating Department foreman and laboratory supervisor before appointment to his present position in 1952. His current work is in the field of corrosion prevention of electroplated products. Previously, he has been engaged in a study of brighteners for electroplating. He is a graduate of Fenn College where he received the B.S. degree in chemical engineering in 1934, and he is a member of the American Electroplaters Society.

- **Willard C. Shaw* and Wilbur C. Crim,** *Delco-Remy Division, Anderson, Indiana, for an Assembling Machine, No. 2,696,321, issued December 7.* This invention relates to a machine for applying tape to electrical coils such as coils for the field windings of electric motors. The machine automatically winds tape around a coil, cuts off the tape, indexes the coil, and applies tape to another portion thereof.

Mr. Crim is general foreman in Delco-Remy's Process Shop. He joined this same Department in 1940 as an apprentice tool maker and was promoted to special machine builder (1949), machine designer (1950), tool engineer in the Jet Blade Program (January 1953), and foreman of the Process Shop (November 1953), to his present position (1954). Mr. Crim served in the Army from 1944 to 1946, separating with the rank of sergeant. This is the first patent resulting from his work with special machinery design.

- **Ralph K. Shewmon and Clyde C. Frey,** *Delco Products Division, Dayton, Ohio, for a Reversible Motor Control, No. 2,696,580, issued December 7.* This patent relates to a control system for a reversible electric motor in which the motor is electrically braked to a stop and is automatically reversed.

Mr. Shewmon is section engineer—fractional motors in Delco Products' Engineering Department. In this capacity, he co-ordinates customer developments as they affect fractional-horsepower motor design. His GM career began in 1930 at GM Radio Corporation where he was employed as a General Motors Institute co-op student. He completed the four-year program in industrial engineering at G.M.I. in 1934. He was transferred in 1932 to Delco Products and later advanced through several positions until appointment to his present post in 1950.

These patent descriptions are informative only and are not intended to define the coverage which is determined by the claims of each one.

Mr. Frey is an engineer on fractional-horsepower controls in the Engineering Department of Delco Products. He came with Delco in 1933, serving first as an inspector and, subsequently, in the Engineering Laboratory, as a motor assembly foreman, and in design and development work (1949). Much of his work has been in the field of motor controls which has resulted in three granted patents. He studied electrical engineering at University of Cincinnati and he is a registered professional engineer.

- **George R. Bayley,** *Buick Motor Division, Flint, Michigan, for an Automobile Heating and Ventilating System, No. 2,696,774, issued December 14.* This patent pertains to an improved construction for conducting incoming air to a heat exchanger of an automobile in which a portion of the automobile frame is employed as an air duct.

Mr. Bayley is associate staff engineer at Buick Motor. He began his career at this Division in 1925 as a co-op student from General Motors Institute. Following his graduation from G.M.I. in 1929, he began work as a student engineer and progressed to body draftsman, chief draftsman, and, since 1945, has held staff-engineering positions related to body and sheet-metal engineering. Among the projects undertaken by Mr. Bayley has been the establishment of a decimal dimensioning system, developed while he was chairman of the GM Standards Subcommittee.

- **Carl W. Blossey,** *Delco Radio Division, Kokomo, Indiana, for an Oscillator Coupling System, No. 2,697,165, issued December 14.* This invention relates to an oscillator coupling system for a radio receiver providing tracking and uniform energy output.

Mr. Blossey is a senior project engineer in the Engineering Department of Delco Radio. He joined the Division as a draftsman in 1936, having previously worked as a foreman for the Crosley Radio Corporation. Since joining the Division, he has been mechanical engineer (1940) and electrical engineer (1944). Currently, he is working in the field of automatic receiver development. A graduate of University of Cincinnati where he received his C.E. degree in 1932, Mr. Blossey also studied at Purdue University Graduate School, 1950-51, where he took courses in advanced radio engineering and pulse techniques.

- **Clarence W. Junge and Carl A. Bierlein,** *Cleveland Diesel Engine Division, Cleveland, Ohio, for a Dual Fuel Engine, No. 2,696,807, issued December 14.* This patent relates to a compression ignition engine operable on gas fuel with pilot oil fuel injected for ignition purposes, or to oil fuel injection alone, with improved governor regulation of the two fuel deliveries.

Mr. Junge is assistant design engineer in the Design Engineering Department of Cleveland Diesel Engine. He joined the Division in 1936 as a layout man in his present Department, became senior layout man the same year, and senior project engineer in 1950. Currently, he is engaged in design work on the 358X engine. Previous projects have related to the design of a dual fuel engine and design of engine controls. Before coming to GM he was associated with the W. S. Tyler Company from 1925 to 1936. He is a co-inventor with W. E. Brill of a vapor compressor for Diesel engines.

Mr. Bierlein is director of the Inspection and Test Department at Cleveland Diesel Engine. His GM career began in 1929 when he was employed by Buick Motor Division as a General Motors Institute co-op student. He holds the B.M.E. degree from G.M.I. (1948). He transferred to Cleveland Diesel Engine Division in 1933 where he has held various positions relating to manufacturing work, including foreman of the Injector Department, superintendent of experimental test, and senior experimental engineer.

- **Paul M. Stivender,** *Cleveland Diesel*

Engine Division, Cleveland, Ohio, for an Automatic Power Plant, No. 2,697,174, issued December 14. This patent relates to an automatic power plant having a Diesel prime mover and an a-c generator and equipped with fully automatic controls for engine starting, speed matching, synchronizing, loading, and shutdown.

Mr. Stivender is assistant chief experimental engineer in the Experimental Engineering Department, Cleveland Diesel Engine. Joining the Division in 1947, he was first a project engineer in the Electrical Engineering Department, became senior project engineer in 1948, and electrical engineer in 1951. He currently is working on the development of a special clutch and previously did work on an automatic Diesel engine control for unattended powerhouses. Previous to joining the Division, he served with Allis Chalmers (1927-31) and Republic Steel (1937-47).

- **William E. Brill* and Allen H. Glasenapp,** *Cleveland Diesel Engine Division, Cleveland, Ohio, for an Engine Accessory Drive, No. 2,698,013, issued December 28.* This covers a heavy duty accessory gear train for an engine. The gear train has torsional flexibility and permits axial movement and angular misalignment of the input and output gears.

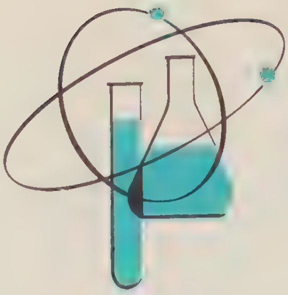
Mr. Glasenapp is no longer with the Division.

- **Gerald H. Fagg and James L. Arthur*,** *Allison Division, Indianapolis, Indiana, for a Thermocouple, No. 2,698,352, issued December 28.* This is a thermocouple with a pair of superimposed plates to support concentric probes and offset terminal posts.

Mr. Fagg is no longer with the Division.

- **Arthur W. Gaubatz,*** *Allison Division, Indianapolis, Indiana, for a Multi-Plate Variable Area Jet Nozzle, No. 2,697,907, issued December 28.* This patent covers a jet nozzle for aircraft propulsion with overlapping flaps or plates adjustable by a ring of tangent screws to vary the nozzle area.

- **Stanford Landell,** *Brown-Lipe-Chapin Division, Syracuse, New York, for a Wheel Cover, No. 2,698,203, issued December 28.* This is a vehicle wheel cover having spaced retaining clips formed with saw-tooth wheel gripping fingers. The teeth of the gripping fingers are alternately angled to provide increased gripping contact.



Mr. Landell is director, Product Engineering and Sales Department of Brown-Lipe-Chapin. He has been in General Motors since 1936 when he joined the GM Art and Color Section as executive engineer. Among his past projects have been improvement of body design and appearance and reduction of body and sheet metal costs for Oldsmobile. Two patents have resulted from his automotive work. He studied electrical engineering at University of Pennsylvania and was associated with Edward G. Budd Manufacturing Company for a ten-year period prior to joining GM. He is vice chairman of the Syracuse Chapter of S.A.E.

- **Dr. Edward J. Martin, Clark E. Quinn, and John Ross,** *Research Laboratories Division, Detroit, Michigan, for a Fuel Measuring System, No. 2,697,939, issued December 28.* This invention relates to a system for measuring fuel consumption of an internal combustion engine that can be used on a car in motion without the necessity of stopping the engine at any time.

Dr. Martin is head of the Physics and Instrumentation Department, Research Laboratories. He joined the Division in 1924 when it was then located in Dayton, Ohio. He received his B.A. degree in 1915, his M.S. degree in 1917, and his Doctorate in 1924—all from University of Michigan. Previous to entering study for his Doctorate he was assistant professor of physics at Case School of Applied Science, Cleveland, Ohio, 1919-22. During World War I he served with both the U.S. Signal Corps and the U.S. Air Corps. He is a member of the Division's 25-Year Club.

Mr. Quinn is research engineer for the Research Laboratories' Physics and Instrumentation Department. He joined the Division in 1942. In 1943 he headed a group for production of electronic instruments and worked in this field until 1953. Much of his work has been concerned with monochrome and color TV microscopy, as well as the production of the Division's breaker amplifier, Sonigage, and link-coupled indicators. He has contributed to three patented inventions in the fields of radio, electronics, and burette fuel metering.

Mr. Ross is supervisor of the Technical Facilities and Services Department, Research Laboratories. He came to the Division in 1929 as a dynamometer operator in the Power Plant Department. He was promoted to dynamometer foreman in 1943 and currently supervises dynamometer facilities of his present Department. His projects have included work on two-cycle engine development, and development of new equipment for GM Technical Center test cells. He is a contributor to a previous patented invention in the field of engine load control.

- **Elmer Olson*,** *Rochester Products Division, Rochester, New York, for a Carburetor, No. 2,698,168, issued December 28.* This is an automatic choke for a carburetor in which the choke valve is controlled by temperature and a suction-operated piston. Means are provided to control the suction effective on the piston by the position of the piston and choke valve for the purpose of varying the rate of movement of the valve in response to suction.

- **Wilgus S. Broffitt,** *Allison Division, Indianapolis, Indiana, for a Thermocouple Mount, No. 2,698,872, issued January 4.* This patent relates to a ring-like thermocouple mount having a plurality of thermocouples mounted at spaced points along its periphery and serving to measure the average temperature of a fluid medium in which the mount is disposed.

Mr. Broffitt is head of the J71 Design Group of Allison's Turbo-Jet Engineering Department. He joined Allison 15 years ago as a junior test engineer and, after advancing through several positions, was promoted to his present post in March 1955. Mr. Broffitt earned the B.S.M.E. degree from University of Kentucky in 1938. Listed among his previous projects is work on the development of the GMR-235 nickel-base alloy for turbine blades and buckets. This is the second patent resulting from his work.

- **Robert L. Camping and Robert C. Treseder,*** *Aeroproducts Operations of Allison Division, Dayton, Ohio, for a Propeller Blade, No. 2,698,666, issued January 4.* This patent relates to a hollow propeller blade, including a thrust member and a camber member brazed thereto, and particularly to an improved tip construction for preventing separation of the members due to water erosion.

Mr. Camping is senior project engineer in Allison's Engineering Department. He joined the Division's Aeroproducts Operations in 1940 as a mathematician and later worked as a project engineer on propeller blades and various propeller components. He was transferred to his present position in 1952 where his work is currently devoted to turbochargers for internal combustion engines. He earned the B.S.E. degrees jointly in mathematics and aeronautical engineering in 1936 and was awarded the M.S. degree in aeronautical engineering in 1937—all from University of Michigan. He serves on several committees of S.A.E. and is a member of I.A.S. and A.S.M.

- **William E. Brill,*** *Cleveland Diesel Engine Division, Cleveland, Ohio, for a Tunnel Crankcase and Crankshaft Mounting Therein, No. 2,698,609, issued January 4.* This relates to crankcase and main bearing supports designed for a vertical multi-row radial engine, providing for endwise insertion of the crankshaft and individual oil scavenging of each row.

- **James S. Burge, Richard M. Goodwin, and Hilton J. McKee,*** *Delco-Remy Division, Anderson, Indiana, for a Rack, No. 2,698,695, issued January 4.* This patent relates to an apparatus for stacking a predetermined number of stator coils to carry them to a successive operation.

Mr. Goodwin is a senior machine designer in the Process Department of Delco-Remy. He was originally employed by Delco-Remy Battery Operations' Design Department in 1937 as a designer. Mr. Goodwin earned the B.S.M.E. degree from Purdue University in 1932. His past projects include work on the design and development of automatic commutator assembly machines and an automatic condenser winding machine.

Mr. Burge is no longer with the Division.

- **Carl E. Hedeen,** *Fisher Body Division, Detroit, Michigan, for a Sunshade Support, No. 2,698,728 issued January 4.* This patent relates to a sunshade support having a ball-and-socket-type joint allowing limited universal movement.

Mr. Hedeen is assistant engineer-in-charge of the Product Engineering Department for two plants of Fisher Body. His current work is with doors and center pillar design for the four-door hard-top body. Previous projects have been related to design supervision of various components, including doors, locks, hinges, check links, window regulators, and curved windshields. His body engineering experience before joining Fisher Body in 1949 was with Chrysler, Murray, and Divco Corporations. A director of the American Society of Body Engineers, he is chairman of the Society's Technical Convention Committee.

- **Paul L. Jones,** *Ternstedt Division, Columbus, Ohio, for a Method of Making Polishing Discs, No. 2,698,783, issued January 4.* This patent relates to a method of making polishing discs in which the gathered fabric at the inner periphery of each disc is heat-formed to fit the arbor and the supporting flanges.

Mr. Jones is resident manager of the Ternstedt plant in Columbus, Ohio, in which capacity he has served since 1949. He joined the Division at Detroit in 1923 as a machine operator and advanced to foreman in 1925, general foreman in 1932, and to Division superintendent in 1934. He was made factory manager of the Ternstedt Trenton, New Jersey, plant in 1937 and in 1945 transferred to the Columbus plant as works manager. He also served as executive assistant, post war planning, in 1945.

- **Leonard J. Mann,** *Frigidaire Division, Dayton, Ohio, for a Refrigerating Apparatus, Including Defrosting Means, No. 2,698,521, issued January 4.* This invention relates to a refrigerator having two separate compartments maintained at different temperatures, and having means for quickly defrosting the evaporator of the high temperature compartment during each off cycle of the compressor.

Mr. Mann is a senior project engineer at Frigidaire where he is engaged in development of household refrigerator units. This is the first patent resulting from his work in this field. He has served in his present capacity since 1953 and formerly his work was concerned with production engineering for refrigerator units. Mr. Mann's entire career has been with Frigidaire, starting in 1931. Later, he was enrolled as a co-op student at University of Cincinnati where he was awarded the M.E. degree in 1940. He served for four years in the U. S. Army separating with the rank of 1st lieutenant. He rejoined Frigidaire in 1946 as a draftsman.

- **Marshall D. McShurley and Donald G. Mahoney,** *Delco-Remy Division, Muncie, Indiana, for a Valve, No. 2,698,630, issued January 4.* This patent deals with a valve for controlling the flow of molten lead in machines for casting battery grids and the like.

Mr. McShurley is general foreman of the Machine, Tool, and Die Room at the Delco Battery Operations of Delco-Remy. This is the third patent resulting from his work in the fields of sheet lead rolling and assembly operations in the battery industry. Mr. McShurley joined Delco-Remy in the Process Department at Anderson, Indiana, in 1927. The following year he transferred to the Muncie plant and was made foreman in 1943. During the same year he was advanced to general foreman and in 1950 was made a senior engineer. He was appointed to his present position in 1954.

Mr. Mahoney is a process engineer in Delco-Remy's Delco Battery Operations Process Department. He joined the Division at Anderson, Indiana, in 1933, becoming a college graduate-in-training in 1934, process engineer in 1936, transferred to Muncie as a junior engineer in 1942, and was promoted to his present position in 1943. A graduate with the M.E. degree from Purdue University in 1933, he has contributed to two patented inventions related to sheet lead rolling and assembly operations in the battery industry.

- **Edwin F. Rossman,** *Delco Products Division, Dayton, Ohio, for a Shock Absorber, No. 2,698,675, issued January 4.* This patent relates to a shock absorber in which the valving for controlling fluid displacement is arranged in a serial stack to gain advantage of a controlled pressure rise.

Mr. Rossman serves as a research engineer in the Engineering Department at Delco Products. Employed by this Division in 1927 as a shock-absorber engineer, he became assistant chief engineer in 1936, chief engineer in 1940, and consulting engineer in 1945. During his career, Mr. Rossman has worked on the development of a ride levelizer, air sus-

These patent descriptions are informative only and are not intended to define the coverage which is determined by the claims of each one.

pension systems, and shock absorbers—all of which have led to more than 40 granted patents. He earned the joint B.M.S.E. degree from Massachusetts Institute of Technology and from Harvard Engineering School in 1920.

• **Edmund J. Adams,** *Ternstedt Division, Detroit, Michigan, for a Pivot Construction, No. 2,698,958, issued January 11.* This is a pivot bearing for automobile sun visors or the like in which tapered journal and sleeve members are adjustable endwise of each other for maintaining a desired degree of friction against turning.

Mr. Adams is a design group leader in the Standards Engineering Department of Ternstedt. His first assignment after joining Ternstedt in 1939 was as detailer in his present Department. Since then he has been layout draftsman (1940), senior layout draftsman (1942), aircraft standards engineer (1943), and automotive standards engineer (1945). He was promoted to his present position in 1952. Mr. Adams is a member of the Engineering Society of Detroit.

• **Max G. Bales,** *Delco-Remy Division, Anderson, Indiana, for a Retainer, No. 2,699,345, issued January 11.* This patent is directed to a packing for the shaft of a distributor structure. The packing maintains the shaft lubricated during operation of the distributor.

Mr. Bales has been a staff engineer assigned to military equipment since 1949. After earning the B.S.E.E. degree from Purdue University in 1933, he was employed by Delco-Remy in its Production Department and in early 1934 he was designated as a student engineer. In 1939 he became a junior engineer and in 1942 was promoted to senior engineer. He is a member of the Society of Automotive Engineers.

• **Anton F. Erickson,** *Moraine Products Division, Dayton, Ohio, for a Multiple Gear Fluid Pump, No. 2,699,122, issued January 11.* This patent relates to a fluid pump having a plurality of pinion gears that are driven from a single drive gear. The axes of the pinion gears are unequally spaced around the periphery of the drive gear to break up the discharge phase relationship.

Mr. Erickson is assistant supervisor of brake research engineering. He was employed by Delco Brake Division in 1937 as a brake engineer. In 1942, when Delco Brake merged with Moraine Products Division, he was made general foreman of the Production Machining Department. In 1944 he was transferred to the position of assistant supervisor of the Engineering Laboratory. In 1954 he was transferred to his present position. This is the tenth patent granted as a result of Mr. Erickson's work. He is a member of the Society of Automotive Engineers.

• **Arthur W. Gaubatz,*** *Allison Division, Indianapolis, Indiana, for a Detachable Combustion Chamber for Gas Turbines, No. 2,699,040, issued January 11.* A combustion chamber for gas turbines including an outer shell, a combustion liner, and a transition section to connect to the turbine nozzle, specially adapted for assembly and disassembly.

• **William G. Livezey,* John E. Storer, Jr.,* and Elmer A. Richards,*** *Allison Division, Indianapolis, Indiana, for a Transmission Control System, No. 2,699,074, issued January 11.* This is a control system for a transmission employing change speed gears and a hydraulic torque converter, together with a lockup clutch. There are means to control engagement of the clutch in response to the relative speed of the torque converter members, and to release the clutch whenever there is a change in the gearing.

• **Archie D. McDuffie,** *Research Laboratories Division, GM Technical Center, Detroit, Michigan, for a Short Flame Travel Combustion Chamber Design, No. 2,699,160, issued January 11.* This patent relates to a combustion chamber in which the burning commences in the middle and extends in opposite directions in cavities formed in the head and the piston.

Mr. McDuffie is an assistant department head of the Research Laboratories' Automotive Engines Department. He began his GM career in 1931 at Cadillac Motor Car Division as a co-op engineering student from General Motors Institute. Upon being graduated from G.M.I. in 1934, he joined the Research Laboratories Division. His recent projects have been concerned with combustion chamber development and high compression engines. He is a member of the Society of Automotive Engineers and the Engine Test Code Subcommittee of the General Technical Committee of GM.

• **Ronald R. Pees,** *Delco Products Division, Dayton, Ohio, for a Control Device, No. 2,699,161, issued January 11.* This is a control device for a Diesel engine that will automatically stop the engine if the pressure in the lubricating system for the engine fails to reach a predetermined safe value after a predetermined period of operation of the engine.

Mr. Pees is manager of Industrial Sales for Delco Products. Employed by the Division since 1935, he has worked as a clerk, in the Engineering Laboratory (1936), in engineering design (1938), as a project engineer (1943-50), assistant section engineer (1950-54), and section engineer (1954). A 1933 graduate of Ohio Northern University with the B.S. degree in electrical engineering, Mr. Pees served from 1943-45 with the Navy Department's Bureau of Ships. This is the first patent resulting from his work.

• **Robert C. Treseder,* Albert P. Dinsmore, Richard E. Moore, Howard M. Geyer,* and Kenneth L. Berninger,** *Aeroproducts Operations of Allison Division, Dayton, Ohio, for a Control for a Plurality of Variable Pitch Counterrotating Propellers, No. 2,699,304, issued January 11.* This patent relates to a variable pitch propeller assembly of the counterrotating type, including a fluid pressure system for adjusting propeller pitch and an electronic governor for controlling the fluid pressure system.

Mr. Dinsmore serves as a project and design engineer in the Engineering Department of Aeroproducts where he is currently engaged in developmental work on supersonic propellers. He originally joined Aeroproducts in 1942 as a vibration tester and was appointed to his present position in 1946. His previous projects have included developmental work on electronic controls. Mr. Dinsmore attended Miami University and was employed by Hamilton Standard Division of the United Aircraft Corporation prior to joining Aeroproducts.

Mr. Berninger serves as senior project engineer in the Engineering Department at Aeroproducts Operations of Allison. His employment dates from his first position as a draftsman at Delco-Remy in 1928. He transferred to Aeroproducts as a designer in 1941 and advanced to his present position in 1943. His projects concern the development of aircraft propellers and accompanying controls. This work has

These patent descriptions are informative only and are not intended to define the coverage which is determined by the claims of each one.

resulted in 14 patents; in addition, previous assignments contributed five patents in automotive and commercial-product design. Mr. Berninger studied electrical engineering at Purdue University.

Mr. Moore is no longer with the Division.

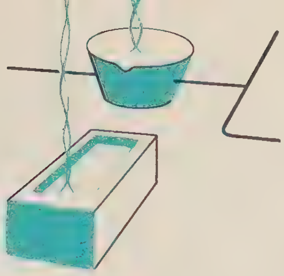
- **Robert Schilling,** *Research Laboratories Division, Detroit, Michigan, for a Wheel or Similar Article, No. D-173,884, issued January 18.* This is an ornamental design for a vehicle wheel.

Mr. Schilling is head of the Engineering Mechanics Department at the Research Laboratories where he specializes in fatigue studies of metals and gear design. He was graduated from Munich Technical University, in his native Germany, as a mechanical engineer in 1922. He came to the United States in 1927, joining the Oldsmobile Division in Lansing, Michigan. There, his work was related to design of engines, knee-action suspension, and automatic transmissions. In 1937 he was transferred to the GM Product Study Section where he remained until shortly after World War II when he joined the Research Laboratories.

- **Edward N. Cole,** *Chevrolet Motor Division, Detroit, Michigan, for a Fluid Seal Damper, No. 2,700,562, issued January 25.* This patent covers a seal for a water pump, the parts of which may be tuned to eliminate seal squeal.

Mr. Cole is chief engineer of Chevrolet. He was plant manager of the Cadillac Cleveland Tank Plant at the time of his transfer to Chevrolet in 1952. A graduate of General Motors Institute with a B.M.E. degree, Mr. Cole was employed in 1933 by Cadillac as a laboratory assistant. He became chief engineer in 1947 and works manager in 1950. Among his projects have been the design of the Cadillac 331 cu in. engine and production of Walker Bulldog tanks. He is co-author of several papers and has participated in various S.A.E. committee activities.

- **Howard M. Geyer,*** *Aeroproducts Operations of Allison Division, Dayton, Ohio, for a Vibration Driven Actuator, No. 2,700,542, issued January 25.* This patent relates to a rotary actuator having a driving member and a driven member, rotation being imparted to the driven member in response to vibrations established by the driving member.



- **John H. Heidorn,** *Frigidaire Division, Dayton, Ohio, for a Refrigerator Door Construction, No. 2,700,195, issued January 25.* In this patent, rigid metal members are connected along the long sides of one panel of a two-panel molded plastic refrigerator freezing compartment door. The ends of the members are wedgingly locked to the short sides of the panel, and the members counteract warping of the panel and maintain the door in alignment with its jamb.

Mr. Heidorn serves as a section engineer at Frigidaire. He joined the Division in 1938 as a stock tracer and was enrolled during the same year as a co-op engineering student at General Motors Institute. He was promoted to layout in 1945, to project engineer in 1948, to senior engineer in 1950, and to his present position, section engineer, in 1953. His major projects have been concerned with compressor development and his work has resulted in two patents in this field.

- **Keith K. Kesling,*** *Frigidaire Division, Dayton, Ohio, for a Sealing Gasket, No. 2,700,197, issued January 25.* This patent relates to a soft rubber-like insert placed in the hollow portion of a rubber door seal at the corners of the door to prevent buckling or collapsing of the seal at the door corner bends.

- **William T. Mears,** *Guide Lamp Division, Anderson, Indiana, for a Universal Mounting Device for Spot Lamps, No. 2,700,525, issued January 25.* A spot lamp mounting device utilizing a bushing which adapts it for use on various makes and models of vehicles, regardless of the precise location of the windshield post, thereby allowing standardization of parts.

Mr. Mears has served as an experimental engineer in the Engineering Department of Guide Lamp since 1945. He first joined Guide Lamp in 1928 as a tool maker. This is the fifth patent resulting from his laboratory work, which has included several years as a foreman in the Model Shop.

- **Clarence E. Morphew and Daniel M. Adams,** *Cadillac Motor Car Division, Detroit, Michigan, for a Headlamp Closure, No. 2,700,725, issued January 25.* A vehicle headlamp mounting wherein the headlamp is recessed into the fender and a hydraulically actuated closure is provided to hide the headlamp from view when it is not in operation.

Mr. Morphew serves as assistant staff engineer in the Body and Experimental Section, Cadillac Motor Car. He joined the Division in 1936 as a detail draftsman and has advanced through various engineering positions related to automotive development and to military tank design. His current major project concerns the styling and engineering of new sheet-metal automotive components. Previously, he worked on the initial development of the panoramic windshield. Mr. Morphew attended Lawrence Institute of Technology and Wayne University, both in Detroit.

Mr. Adams serves as staff engineer in the Body and Sheetmetal Section of Cadillac Motor Car. In 1934 he joined Cadillac as a clerk in the Inspection Department and, prior to promotion to his present post (1953), he served as a draftsman, assistant section engineer—sheetmetal (Detroit plant), and staff engineer—armament and turret (Cleveland plant). Mr. Adams attended Detroit Institute of Technology and University of Michigan. He is a member of the S.A.E., American Society of Body Engineers, and the Engineering Society of Detroit.

- **Arthur V. Somers,** *AC Spark Plug Division, Flint, Michigan, for a Roller Machine and Method for Impregnating Web Material, No. 2,700,620, issued January 25.* This invention relates to a roller impregnating machine and a method for partially impregnating filter paper with resin without filling the pores to increase the wet strength of the paper and retain the proper degree of porosity for use in filters.

Mr. Somers is a senior experimental chemist in the Research and Development Laboratory at AC Spark Plug. He has served in this capacity since 1950, following advancement through several positions in the Laboratory. He was employed by the Division as a student engineer in 1940, following his graduation from Michigan State College with the B.S. degree in chemistry. During World War II, he was in charge of the chemistry laboratory for Browning machine gun production. Other projects have related to development of electroformed paint spray masks, development of resin and impregnators used in paper oil filters, and development of rubber molded insulators.

A Typical Problem in Engineering:

Determine the Shear Loads and Bending Moments in a Statically Indeterminate Beam

By HAROLD E. HELMS
Allison
Division

Moments and shear loads guide the selection of beam shape and material

In the analysis of forces acting upon component members of a structure, the calculations necessary to arrive at the stresses to which each member is subjected can be relatively simple or very involved. In the case of determining the stress in a beam which is statically indeterminate, it is not possible to compute all external reactions and resisting moments by the simple application of the fundamental equations of statics. A more detailed analysis is required.

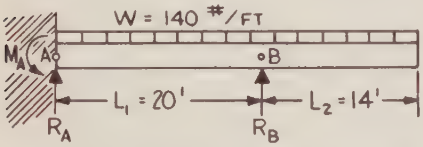


Fig. 1—The determination of the shear loads and bending moments of this statically indeterminate beam depends upon the successful application of fundamental statics and strength of materials to first find the resisting moment M_A and then the external reactions R_A and R_B.

AN ESSENTIAL requirement for maintaining the weight of an aircraft engine compatible with the aircraft's overall design is the proper selection of material and the geometry of structural parts. This requirement also applies to the design of aircraft engine test-stand facilities.

In order to satisfy this essential requirement it is necessary that the design engineer make an analysis to determine the loads which act upon the different structural members. When the loads have been determined it is possible to properly select suitable structural materials and, in turn, proportion the structural members to withstand the loads.

The problem to be presented is a typical one which frequently must be analyzed by Allison Division's engineers in the course of selecting the proper material and geometry of structural parts for the design of an aircraft engine test stand. It is a problem dealing with a beam which is statically indeterminate. The method of analysis for a beam of this type is slightly more rigorous than for a beam which is statically determinate, when it is then possible to compute all external reactions and resisting moments by using the basic equations of statics:

$$\Sigma M = 0, \Sigma H = 0, \text{ and } \Sigma V = 0.$$

Problem

An aircraft engine test stand exhaust stack is loaded as a beam fixed at one end with a simple support part way up the stack. The stack or "beam" must be capable of carrying a uniform air load of 140 lb per ft due to a 100-mph wind. Fig. 1 represents the beam (shown in a horizontal instead of its vertical position) and the location of its support.

The problem is to determine the shear loads and bending moments in this beam, which is statically indeterminate. The stresses due to shear and bending are not to be calculated. The problem's solution must include the shear diagram and bending-moment diagram. The modulus of elasticity E and the moment of inertia I of the beam are to be constant throughout its length.

It will first be necessary to determine the unknown moment M_A acting on the fixed end of the beam. The calculation of this moment will make the beam statically determinate. The solution to the problem then may be completed by the application of the basic equations of static equilibrium.

The solution which will appear in the July-August 1955 issue of the GENERAL MOTORS ENGINEERING JOURNAL will be based upon the area-moment method and, specifically, the second area-moment proposition.

Solution to the Previous Problem

When a jet aircraft undergoes a yawing flight maneuver, a moment of force is produced about the aircraft's vertical axis. This moment of force, which is applied to the jet engine's turbine shaft, is due to the gyroscopic action of the spinning turbine wheels and has a direct bearing on the bending stress to which the shaft is subjected. The bending stress combined with the shearing stress in the shaft due to the torque or bending moment produces a maximum tensile stress of 37,915 psi which is concentrated at one of the shaft's bearings. This is the solution to the problem presented in the March-April 1955 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

THE first step in the solution to the problem is concerned with determining the torsional shearing stress in the hollow cylindrical turbine shaft having over-hung turbine wheels (Fig. 1).

The turbine shaft is being used for the transmission of power. The twisting moment or torque T, therefore, which is applied to the shaft, may be represented by the following formula:

$$T = 63{,}000\,(hp)/N \qquad (1)$$

where

T = twisting moment or torque applied to the turbine shaft (in.-lb)

hp = horsepower transmitted by the turbine shaft

N = turbine shaft speed (rpm).

The problem stated that the jet engine develops 16,560 hp at a turbine shaft speed of 11,000 rpm. If the values for horsepower and shaft speed are substituted into equation (1), the torque or twisting moment applied to the turbine shaft will be:

$$T = 63{,}000(16{,}560)/11{,}000$$

$$T = 94{,}844 \text{ in.-lb.}$$

The maximum shearing stress S_S in the turbine shaft due to the torque of

Determine the Maximum Tensile Stress in a Jet Aircraft Turbine Shaft with Over-hung Turbine Wheels

By ROBERT W. LEWIS
Allison
Division

Maximum tensile stress a function of combined shear and bending

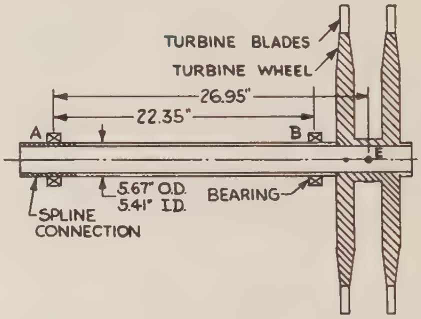


Fig. 1—A jet engine turbine shaft with over-hung turbine wheels is subjected to a bending moment of force when the aircraft undergoes a yawing flight maneuver. This moment of force is caused by the gyroscopic action of the spinning turbine wheels.

twisting moment T can be expressed by the following relationship:

$$S_s = Tc/J \qquad (2)$$

where

S_s = shearing stress at the extreme outer fibers of the shaft (psi)

c = distance from the shaft's axis to the extreme outer fiber (in.)

J = polar moment of inertia of the hollow turbine shaft (in.^{4}).

The polar moment of inertia J of the hollow turbine shaft having an outside diameter d_o of 5.67 in. and an inside diameter d_i of 5.41 in. can be expressed by the following equation:

$$J = \pi(d_o^4 - d_i^4)/32. \qquad (3)$$

Substituting the values for the inside and outside diameters of the turbine shaft into equation (3) results in a turbine shaft polar moment of inertia of:

$$J = \pi(5.67^4 - 5.41^4)/32$$

$$J = 17.36 \text{ in.}^4.$$

The maximum shearing stress due to the torque or twisting moment will occur at the extreme outer fiber of the shaft or at a distance c of 2.83 in. from the shaft's axis. Substituting this value for c and the previously calculated values for T and J into equation (2) gives the following value for the shearing stress S_s in the turbine shaft due to torsion:

$$S_s = 94{,}844(2.83)/17.36$$

$$S_s = 15{,}461 \text{ psi}.$$

When the aircraft goes through the yawing flight maneuver (turns horizontally about its vertical axis) a moment of force is produced about that axis which is caused by the gyroscopic action of the spinning turbine wheels. This gyroscopic moment of force must be calculated next because of its effect on the bending stress in the tubrine shaft.

The gyroscopic moment of force equation, as expressed in the statement of the problem, was:

$$M_g = (\omega\Omega I_p)12/g \qquad (4)$$

where

M_g = moment of force applied to the turbine shaft due to the gyroscopic action of the spinning turbine wheels when the aircraft yaws (in.-lb)

ω = angular velocity of the turbine shaft (radians per second)

Ω = yaw velocity of the aircraft (radians per second)

I_p = polar mass moment of inertia of the turbine shaft (lb-ft^2)

g = gravitational conversion factor (32.2 ft/sec^2).

Data presented in the statement of the problem established that the yaw velocity Ω of the aircraft about its vertical axis was 3.5 radians per second and that the polar mass moment of inertia I_p of each turbine wheel was 29.66 lb-ft^2. The only unknown, therefore, in equation (4) which has to be determined before solving for M_g is the angular velocity ω of the turbine shaft.

The relationship between the angular velocity ω, in radians per second, and the speed N, in revolutions per minute, of the turbine shaft can be expressed by the following equation:

$$\omega = 2\pi N/60. \qquad (5)$$

The speed of the turbine shaft was stated as 11,000 rpm. Substituting this value for N into equation (5) gives the following value for the shaft's angular velocity:

$$\omega = 2\pi(11{,}000)/60$$

$$\omega = 1{,}152 \text{ radians per second.}$$

The moment of force applied to the turbine shaft due to the gyroscopic action of the spinning turbine wheels when the aircraft yaws can now be determined by substituting into equation (4) the known values for ω, Ω, and I_p.

$$M_g = 1{,}152(3.5)2(29.66)12/32.2$$

$$M_g = 89{,}135 \text{ in.-lb.}$$

The weight of each turbine wheel was established in the statement of the problem as 85 lb. It also was established that the aircraft was under a vertical flight maneuver load of 10 G's which, in reality, is a load 10 times the force of gravity. This would then make the actual weight of the two turbine wheels on the shaft equal to 2(85) (10) or 1,700 lb.

Fig. 2 shows a free-body diagram of the turbine shaft under consideration and the direction of forces which are acting upon the shaft. It can be seen that a reaction force R_1 is acting downward at bearing A and an upward reaction force R_2 acting at bearing B. Also shown is the 89,135 in.-lb moment of force applied to the shaft because of the spinning action of the turbine wheels when the aircraft is yawing. The 1,700 lb total weight of the turbine wheels has been equally distributed at points C and D which represent the mid-point of the

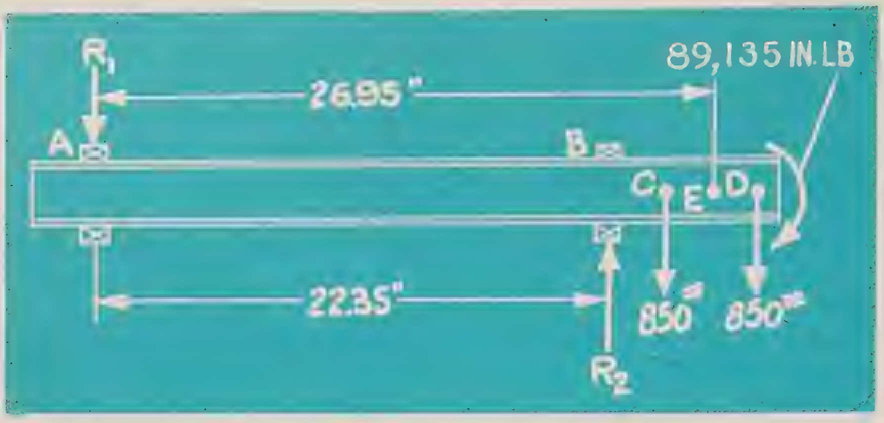


Fig. 2—Free-body diagram of the turbine shaft. There are two reaction forces R_1 and R_2 acting upon the shaft's bearings *A* and *B*. The shaft is also subjected to a moment of force of 89,135 in.-lb caused by the gyroscopic action of the spinning turbine wheels when the aircraft yaws. The points ***C*** and ***D*** represent the mid-point of each turbine wheel width. The weight of each turbine wheel on the shaft is 850 lb when the vertical flight maneuver load of 10 *G*'s is taken into account. The point ***E*** represents the mid-point between *C* and *D* where the combined weight of the turbine wheels can be considered to be located.

width of each turbine wheel.

The reactions R_1 and R_2 can be determined by writing a moment equation about point *E* which can be considered to be the point where the total weight of the turbine wheels on the shaft is located. If all clockwise moments are considered positive the following moment equation about point *E* can be written:

$$\Sigma M_E = 0$$

$$89{,}135 + 4.60R_2 - 26.95R_1 = 0.$$

From a vertical force resolution:

$$R_2 = R_1 + 1{,}700.$$

Substituting $R_2 = R_1 + 1{,}700$ into the above moment equation gives:

$$89{,}135+4.60(R_1+1{,}700)-26.95R_1=0$$

$$89{,}135+4.60R_1+7{,}820-26.95R_1=0$$

$$22.35R_1 = 96{,}955$$

$$R_1 = 96{,}955/22.35$$

$$R_1 = 4{,}338 \text{ lb.}$$

The reaction force R_2, therefore, will equal:

$$R_2 = R_1 + 1{,}700$$

$$R_2 = 6{,}038 \text{ lb.}$$

The shear at all sections of the shaft can be represented by the shear diagram shown in Fig. 3a. The bending moment at any section of the shaft can be represented by the moment diagram shown in Fig. 3b. It can be seen that the shaft's dangerous section is located at the mid-point of bearing *B* where the shear becomes zero. The maximum bending moment which occurs can be found by multiplying the forces to the left of the mid-point of bearing *B* by their lever arms. In this case, there is only one force—the reaction force R_1. The maximum bending moment *M* in inch-pounds in the turbine shaft is:

$$M = 4{,}338(22.35) = 96{,}954 \text{ in.-lb.}$$

The maximum bending stress S_b due to the bending moment at the mid-point of bearing *B* can be expressed as:

$$S_b = Mc/I$$

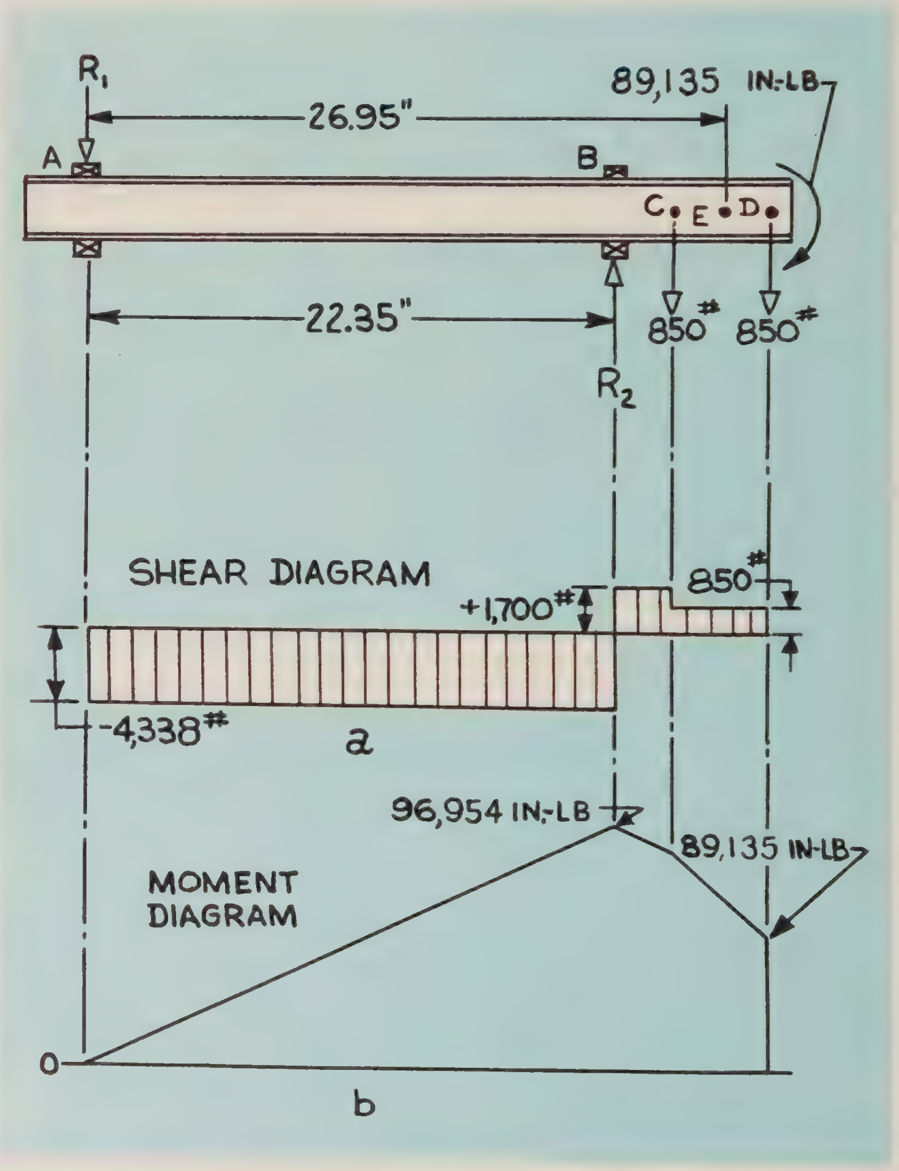


Fig. 3—The shear diagram, (*a*), points out that the dangerous section of the shaft is at the mid-point of the bearing *B* where the shear becomes zero. This is also the point where the maximum bending moment is located as shown in the moment diagram (***b***).

where

S_b = maximum bending stress at the mid-point of bearing B (psi)

M = bending moment at the mid-point of bearing B (96,954 in.-lb)

c = distance from axis of shaft to extreme outer fiber (2.83 in.)

I = moment of inertia of the hollow shaft with respect to the neutral axis (in.^{4}).

The moment of inertia I for the hollow turbine shaft can be expressed as $J/2$. The polar moment of inertia J for the turbine shaft was previously calculated to be 17.36 in.^{4}. The moment of inertia I, therefore, for the turbine shaft is 17.36/2 or 8.68 in.^{4}. Substituting the known values for M, c, and I into equation (6) gives the following value for the maximum bending stress:

$$S_b = 96{,}954(2.83)/8.68$$

$$S_b = 31{,}610 \text{ psi.}$$

The turbine shaft is subjected to the combined effect of bending moment and torque. The fibers at the dangerous section of the shaft, at the mid-point of bearing B, which are farthest from the neutral axis are subjected to the combined effect of the maximum bending stress and the maximum shearing stress. This combined effect is larger than the summation of the previously calculated values for S_b the maximum bending stress and S_s the maximum shearing stress. In order to determine accurately the maximum tensile stress S_t due to the combined effect of the bending stress S_b and the shearing stress S_s which results from torsion at the extreme outer fibers of the shaft at the mid-point of bearing B, it is necessary to use the following strength of materials formula:

$$S_{t\,max} = S_b/2 + \sqrt{S_b^2/4 + S_s^2}. \qquad (7)$$

Substituting into equation (7) the previously calculated values for S_b and S_s gives:

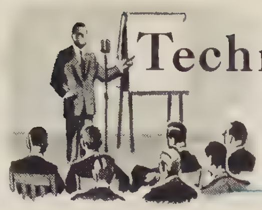
$$S_{t\,max} = 31{,}610/2 + \sqrt{31{,}610^2/4 + 15{,}461^2}$$

$$S_{t\,max} = 15{,}805 + \sqrt{488{,}840{,}546}$$

$$S_{t\,max} = 15{,}805 + 22{,}110$$

$$S_{t\,max} = 37{,}915 \text{ psi.}$$

The solution to the problem, therefore, of determining the maximum tensile stress in the hollow turbine shaft of a jet aircraft undergoing a yawing flight maneuver is 37,915 psi, or almost 19 tons per sq in.



Technical Presentations By GM Engineers

One of the most interesting and profitable means of exchanging engineering and technical ideas is through the speaking engagement. "Talking things over" at meetings of national and local engineering societies, student classes, college forums, as well as some non-technical civic organizations provides opportunities for both the dissemination of engineering knowledge and the stimulation afforded by personal contacts. GM's speaking engineers meet with many audiences and cover a wide variety of subjects, representing many kinds of scientific and technical work carried on in the organization. The presentations made and mutual interests shared by speakers and groups make these meetings an important phase of engineering activity. Listed below are some recent speaking engagements of GM engineers. Requests may be directed to the Educational Relations Section for assistance in obtaining the services of GM engineers to speak before engineering classes or other student groups.

Manufacturing Processes and Facilities

"Rust Prevention in Manufacturing Operations" was outlined by **C. R. Gillette**, chief chemist, New Departure Division, at the January 19 meeting of the American Society of Lubrication Engineers in Meriden, Connecticut.

John J. Donahue, general foreman in the Plant Sanitation Department of the Research Laboratories Division, described "Organized Sanitation at the Research Laboratories" during the Plant Maintenance and Engineering Conference held in Chicago on January 24.

"The Two-Way Road" was the subject of **Lewis E. Cummings**' speech at the National Association of Machine Accountants meeting in Dayton, Ohio, February 10. Mr. Cummings is assistant manager in the Production and Material Control Department of Delco Products Division.

Robert C. Appenzeller, senior engineer in the Electrical Process Department of Delco Products, served as a member of a welding quiz panel before the Dayton, Ohio, Engineers Club, January 11.

Delco Products was also represented at the 3rd Annual Miami Valley Safety Conference held at Dayton, Ohio, February 9. **Fred E. Lewis**, safety engineer, served as co-chairman of the Material Handling Conference session; **Claude M. Willis**, safety director, as co-chairman of the Manufacturers Conference session; **Charles Van Dine**, superintendent of shipping, as co-chairman of the Safety Films Section; and **Floyd O. Denlinger**, safety engineer, as co-chairman of the Facilities and Materials Section.

John W. Armstrong, superintendent of tool engineering, Rochester Products Division, spoke on "Control of Cyanides in Industry" before members of the Industrial Medical and Industrial Hygienists Society at Rochester, New York, February 5.

Robert B. Colten, senior project engineer, Electronics Laboratory, Process Development Section, GM Manufacturing Staff, discussed "Electronics As a Manufacturing Tool" at a meeting of the Bloomfield Manufacturers Club held at Glen Ridge Country Club, Bloomfield, New Jersey, on March 1.

Lester F. Campbell, assistant chief engineer, Electro-Motive Division, participated in the February 11 session of Purdue University's Executive Management Seminar by introducing the subject, "Managing the Engineering Function," and acting as moderator in the ensuing discussion.

A meeting of the Scientech Club of Indianapolis held January 24 at Indianapolis, Indiana, was the occasion for a talk by **Kenneth B. Bly**, supervisor, Product Engineering Department, Light Alloy, Fabricast Division. Mr. Bly's subject was titled, "Production of Aluminum Alloy Torque Converter Castings."

"Production Brazing" was the subject of **Ray Wirt's** talk before the Midwest Welding Conference held in Chicago, Illinois, February 9. Mr. Wirt is welding engineer in the Process Department at Delco-Remy.

Arthur Bender, quality control engineer, Engineering Department, Delco-Remy, discussed "Fundamentals" relating to the subject of quality control before members of the Cummins Engine Company, Columbus, Ohio, on February 17.

Speaking before the Cleveland Engineering Society's 12th Annual Machine Design Conference held at Cleveland, Ohio, February 7, **Donald W. Bain,** chief body draftsman, Body Drafting Group, Chevrolet Motor Division, outlined the subject, "Drafting and Its Relation to Production Requirements."

Research and Testing Equipment and Procedures

The Conference on Analytical Chemistry and Applied Spectroscopy held at Pittsburgh, Pennsylvania, February 28 to March 4, was the occasion for the presentation of several talks by members of the Research Laboratories Division's Physics and Instrumentation Department. **Thomas P. Schreiber,** research physicist, and **Phillip V. Mohan,** junior physicist, made a presentation on "Spectrographic Analysis of a Nickel Base High Temperature Alloy;" **Robert C. Frank,** research physicist, discussed "Homogeneity Study of NBS-GMC Zinc Base Die Cast Alloy Standards;" **David L. Fry,** supervisor, outlined the subjects, "A Cooperative Study on Universal, Semi-Quantitative Spectrochemical Methods," "General Method of Spectrochemical Semi-Quantitative Analysis," and "Spectrographic Analysis of Oils." Mr. Fry, together with **John J. Schultz,** production mechanic, and **Harmon D. Nine,** college graduate-in-training, also demonstrated the "Development of an Accurate and Rapid Method for the Direct Spectrochemical Determination of Metals in Oil."

Participating in the same Conference, **Maurice D. Cooper,** assistant head, Chemistry Department, Research Laboratories, spoke on "A Brief History of the NBS-GMC Base Spectrographic Standards Program." Mr. Cooper also made a joint presentation with **Robert E. Kohn,** research chemist, Research Laboratories, on "Chemical Analysis of NBS-GMC Zinc Base Standards for Manganese, Nickel, and Chromium." **John S. Wolfe,** chief chemist, Engineering Laboratory, Delco Products Division, discussed the "Theory and Design of Spectro Computers for Emission Analysis."

David C. Apps, head, Noise and Vibration Laboratory, GM Proving Ground Section, spoke before the Passenger Car Body and Materials Meeting of the S.A.E. at Detroit, March 1. The subject of Mr. Apps' talk was "Tire Thump, Its Mechanism and Measurement."

Speaking before a recent meeting of the Men's Club of the Clarkston Methodist Church, Clarkston, Michigan, **Richard O. Painter,** assistant head, Experimental Engineering Department, General Motors Proving Ground Section, outlined the subject of "Proving Ground High Speed Photographic Instrumentation and Applications."

"Testing Procedures Used on Various AC Products" was the subject of a talk presented to the West Flint Kiwanis Club, February 22, by **Dr. Robert Smith,** supervisor of physics and metals laboratories, Spark Plug Engineering Department, AC Spark Plug Division.

Automotive Engines

"The New Chevrolet V-8 Engine" was the title of talks given by Chevrolet Motor Division engineers **J. T. Rausch, R. F. Sanders,** and **M. M. Roensch.** Mr. Rausch, engine engineer, Engine Design Group, addressed the S.A.E. Student Chapters of Case Institute of Technology, Cleveland, Ohio, and the University of Michigan, Ann Arbor, Michigan, on February 8 and February 23, respectively. Mr. Sanders, chief passenger car chassis design engineer, Engine and Passenger Car Chassis Department, made his talks before the S.A.E. in Muskegon, Michigan, on January 18 and also before the S.A.E. Pacific Coast Section at San Francisco, California, on February 23. Mr. Roensch, director of laboratory tests, Experimental Engineering Department, appeared before a similar group at Williamsport, Pennsylvania, March 7.

Robert F. Thomson, head, and **George H. Robinson,** senior research metallurgist, Metallurgy Department, Research Laboratories Division, collaborated with **George K. Malone,** senior research chemist, Research Laboratories, and **Eugene B. Etchells,** design engineer, Engineering Department, Chevrolet Motor Division, in the presentation of "The Interrelationship of Design, Lubrication and Metallurgy in Cam and Tappet Performance" before the March 1 meeting of the S.A.E. held at the Hotel Statler, Detroit, Michigan.

The February 25 meeting of the Milwaukee Section of the S.A.E., Milwaukee, Wisconsin, was the occasion for a discussion of "The Effects of Combustion Chamber Shapes on Smoothness of Power" by **Harvey L. Mantey,** laboratory supervisor, Engineering Laboratory, Cadillac Motor Car Division.

Buses, Locomotives, and Off-the-Road Vehicles

Floyd A. Franklin, coach engineer, Engineering Department, GMC Truck and Coach Division, discussed the subject of "Two Level Buses" before S.A.E. Section meetings held at New York on January 19, at Detroit on February 7, and at Philadelphia on February 9.

Speaking before members of the Geography Club of the University of Western Ontario, **Mr. W. G. Goodeve,** administrative engineer, Product Engineering Department, GM Diesel, Ltd., outlined "Locomotive Design Problems Caused by Geographical Location." The Club's meeting was held at London, Ontario, on February 16.

"Modern Earthmoving Equipment" was the topic outlined by **Alan S. McClimon,** manager, Sales Development Section, Euclid Division, before the Boston Chapter of the S.A.E. held at the M.I.T. Faculty Club, February 1.

Electronics

M. J. Manahan, section head, Engineering Department, Delco Radio Division, gave a talk before the radio and electrical engineering student body of Tri-State College at Angola, Indiana, March 7. The subject of Mr. Manahan's talk was "Some Fundamentals of Transistors."

The February 16 meeting of the Noblesville Kiwanis Club, Noblesville, Indiana, was the occasion for a talk on the "Development of the Guide Autronic Eye Automatic Head Lamp Control" given by **Gerald R. Broshar,** field engineer, Engineering Laboratory, Guide Lamp Division.

Automotive Bodies and Chassis

On January 13 **Dr. Gerald M. Rassweiler,** assistant head of the Physics and Instrumentation Department, Research Laboratories Division, described "Co-operative Road Tests of Night Visibility Through Heat Absorbing Glass" to the Highway Research Board meeting in Washington, D. C.

On January 18 **C. W. Lincoln,** chief engineer of Saginaw Steering Gear Division, addressed the Michigan State College Student Branch of the S.A.E. in East Lansing, Michigan. He spoke on "Power Steering."

Before a meeting of the Rotary Club of Lapeer, Michigan, on January 25, Mr. Lincoln outlined "The Development of the Recirculating Ball Steering Gear."

Philip O. Johnson, engineer-in-charge of the Test Coordination and Proving Ground Department of Fisher Body Division, presented a talk before the S.A.E.'s Golden Anniversary Passenger Car, Body, and Materials meeting which was held in Detroit on March 2. Mr. Johnson's subject was "Automobile Body Loads."

Aircraft Engines

Addressing the Institute of Radio Engineers at Dallas, Texas, February 10, **Karl F. Wacker,** experimental engineer, Electrical and Parts Test Instrumentation Group, Aircraft Engine Operations, Allison Division, described "Instrumentation Systems for Jet Engine Testing."

P. N. Bright and **E. M. Howard,** also of Allison Division, spoke to professional and student groups at Lafayette, Indiana, and Ann Arbor, Michigan. Mr. Bright, group project engineer, Stress Analysis and Weight Control, Power Turbine Engineering, talked on "Structural Design Problems of Gas Turbine Engines" before the Society for Experimental Stress Analysis at Purdue University, March 7; Mr. Howard, test project control group engineer, Aircraft Engines Operations, chose "Aircraft Engine Testing" for the subject of his talk, and outlined a basic philosophy of testing and its application to engine development. The talk was given on March 16, before the student branches of the American Society of Mechanical Engineers and S.A.E. of the University of Michigan.

Lubrication and Bearings

L. D. Cobb, manager of the Research and Development Laboratory, New Departure, lectured at a meeting of the American Society of Lubrication Engineers in Philadelphia, Pennsylvania, on January 27. His topic was "Bearing Seals—Design, Application, and Limitations."

The meeting of the Association of Lubrication Engineer's Ontario Chapter, held at Toronto on February 1, was the occasion for a talk on "Diesel Engine Lubrication" given by **D. M. Jenkins** of GM Diesel, Ltd. Mr. Jenkins is chief engineer in the Product Engineering Department.

"Antifriction Bearing Lubrication" was the subject of **C. R. Gillette's** talk given at a lubrication course sponsored by the A.S.L.E. on February 14 at the Franklin Institute, Philadelphia, Pennsylvania. Mr. Gillette is chief chemist at New Departure Division.

"Powder Metallurgy" was the title of a paper presented January 1 and February 22 by **Roland P. Koehring,** chief metallurgist, Engineering Department, Moraine Products Division. The talks were given before the American Ordnance Association, Watertown Arsenal, in Boston, Massachusetts, on January 27 and at a meeting of the Society for the Advancement of Management held in Dayton, Ohio, on February 22.

Power

Kendall O. Bower, a-c controls engineer, and **E. A. Armstrong,** consultant, Industrial Sales Department—both of Electro-Motive Division—presented a joint paper before the American Power Conference, held March 31 in Chicago, Illinois. The title of the paper was "Conception, Development and Application of Mobile Electric Automatic Generating Equipment for Utilities."

Herman Esch, chief engineer of the power plant, Elyria plant of Brown-Lipe-Chapin Division, was the principal speaker at the annual meeting of the Central West Virginia Mining Institute, held February 4 in Clarksburg, West Virginia. His subject was "Types of Coal for Spreader Stokers."

Styling

G. T. Christiansen, administrative assistant, General Motors Styling Section, discussed "Styling Today's Car" in a talk given before the Dayton Section of the S.A.E., Dayton, Ohio, February 17. Speaking in a similar vein, **Charles M. Jordan,** designer, spoke at a meeting of the Junior Section of the S.A.E., Detroit, Michigan, March 7. The subject of Mr. Jordan's talk was "Styling is More Than Skin Deep."

Ceramic Engineering

The Empire Hotel, New York City, was the scene of **Karl Schwartzwalder's** talk on "Compression Injection and Isostatic Molding," presented January 21 before the Metropolitan New York Section of the American Ceramic Society. Mr. Schwartzwalder is director of research in the Spark Plug Engineering Department, AC Spark Plug Division.

General Engineering

Dr. Gerald M. Rassweiler, assistant head of the Physics and Instrumentation Department, Research Laboratories Division, spoke on "The Life History of an Equation" before the Industrial Mathematics Society. The meeting was held on January 20 at the General Motors Technical Center.

"Man's Brain," was the title of an address by **L. J. Mathiason,** given before the Rotary Club of Crawfordsville, Indiana, February 2. Mr. Mathiason is an engineer with the Calculation Group, Engineering Services, Aircraft Engine Operations, Allison Division. The use of high speed computers in engineering computations and for various phases of management was the subject of his address.

"The Engineer in Design and Development" was the title of a talk presented to senior mechanical engineering students at Michigan State College, East Lansing, Michigan, February 15. The speaker was **John Dolza,** engineer in charge of the Power Development Group, Engineering Staff, General Motors Technical Center.

"The Invisible Man, The Engineer" was the title of an address given February 22 and 24 by two AC Spark Plug Division engineers. **Leonard E. A. Batz,** senior project engineer, Engineering Department, presented it before the Flint Shrine Club, Flint, Michigan, and **Martin J. Caserio,** chief engineer, Automotive Products, Engineering Department, appeared on a panel which discussed the subject over Channel 5, WNEM-TV, Saginaw, Michigan. "National Engineers Week" was the title of a talk given by **Robert W. Chase,** systems engineer, Defense Engineering Department, AC Spark Plug, before the Optimist Club of Flint, Michigan, February 2.

Eric R. Brater, assistant chief engineer, Cleveland Diesel Engine Division, was principal speaker at the Cleveland, Ohio, Section meeting of the S.A.E. held February 14. His topic: "The Amazing Engineering and Architecture of the Ancient Egyptians."

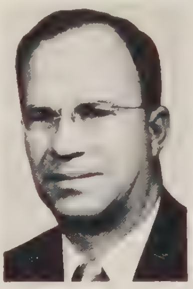
"The University of Michigan Engineering School Expansion" was the subject of an address given February 28 by **Walter B. Herndon,** works manager, Manufacturing Department, Detroit Transmission Division. Mr. Herndon spoke before business and industrial leaders of Battle Creek, Michigan.

Correction to the Previous Solution

Attention has been called to two errors which appeared in the January-February 1955 GENERAL MOTORS ENGINEERING JOURNAL solution to the problem "Find the Wheel Loads of a Railway Truck of Novel Design." The first error appeared in STEP 1 of the calculations appearing on page 59. The relationship $ab:AB$ as $bC:AC$ should be corrected to read $aB:AB$ as $bC:AC$. This same correction for aB in place of ab should be made for the equation immediately following the above relationship and should read $aB = (AB)(bC)/AC$.

The second error appeared in STEP 5 of the calculations appearing on page 59. In solving the moment equation for jx, the substitution for W_1 and W_2 should be 25,000 instead of 20,000.

Contributors to May-June 1955 Issue of GENERAL MOTORS *Engineering* JOURNAL



WALTER T. BURWELL,

contributor of "How Industrial Suppliers and Design Engineers Work Together," serves as director of the Special Vehicle Engineering Group in Chevrolet Motor Division's Central Office, Detroit, Michigan.

The Special Vehicle Engineering Group is responsible for the design instigation of factory installed optional equipment, including specially requested options, and for maintaining liaison between the Chevrolet Engineering Department and other General Motors Divisions and sections interested in these matters. Another part of the Group's duties is to create special designs for passenger car and truck fleet operators and government agencies and to design tactical vehicles used by the military services.

Mr. Burwell joined Chevrolet in 1941 as a project engineer at the Chevrolet Aviation Engine Division in New York. Four years later he was transferred to Chevrolet Central Office as a senior project engineer, dealing with problems of carburetion. Progressing through the positions of design engineer and assistant staff engineer in charge of engine design, he was promoted in February 1952 to staff engineer in charge of passenger car chassis design. He assumed his present duties in August 1953.

University of Michigan granted Mr. Burwell the B.S.M.E. degree in 1926. Prior to joining General Motors, he was assistant chief engineer at the Tillotson Manufacturing Company of Toledo, Ohio.

Mr. Burwell is a member of the Detroit Section of the Society of Automotive Engineers and of the American Ordnance Association. He has served both in the Motor Fuel Division and on the Fuel Volatility Panel of the Coordinating Research Council.

MERRILL G. HINTON, JR.,

co-contributor of "Analyzing a Typical, Simple-Cycle Turbo-Jet Design and Development Program," is a project engineer in the Turbo-Jet Design Group at Allison Division, Indianapolis, Indiana. He has served in this capacity since early 1952, and most of his current work centers around fabricated parts design, principally sheet metal and weldments.

After service with the U. S. Navy for two years, Mr. Hinton joined Allison Division in 1947, enrolling as a cooperative student in the mechanical engineering program at General Motors Institute, Flint, Michigan. As he gained experience at Allison, he progressed through different levels of design responsibility in the turbo-jet design program. In 1951 he became a detail engineer and in early 1952 was advanced to his present position. General Motors Institute granted him the Bachelor of Mechanical Engineering degree in 1952 and elected him to Alpha Tau Iota honorary society.

Included among Mr. Hinton's duties as project engineer has been work on advanced engine design and stress and weight analysis, as well as concentrated design work in the turbo-jet development program which forms the basis of the paper appearing in this issue.

J. RALPH HOLMES,

contributor of "Development of an Automobile Air Conditioning System for Underhood Installation," is chief engineer of Harrison Radiator Division, Lockport, New York. His paper presented in this issue is an adaptation of "Air Conditioning the Automobile," a paper which won for him the American Society of

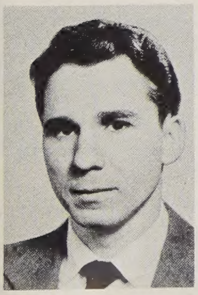
Refrigerating Engineers Best Section's Paper Award for 1954.

Mr. Holmes, who received his engineering education in Cleveland, Ohio, joined the engineering staff of Harrison Radiator in 1928. Four years later he was promoted to his present position. As chief engineer, Mr. Holmes has actively directed the development and design improvements of Harrison Radiator products which include automotive radiators, thermostats, and air conditioning systems and heat exchangers for various applications.

Prior to joining Harrison Radiator, Mr. Holmes was associated with the Goodyear Tire and Rubber Company, Akron, Ohio, as a designer; the Cleveland Tractor Company, Cleveland, Ohio, as an experimental engineer; the Wheeler Schebler Carburetor Company, Indianapolis, Indiana, as a sales engineer; and the Chandler Motor Car Company, Cleveland, Ohio, as assistant chief engineer.

The technical affiliations and committee memberships of Mr. Holmes are many. He is a member of the Society of Automotive Engineers and serves as the chairman of that Society's Motor Vehicle Air Cooling Committee. He also is a member of the Radiator Standardization Working Committee No. 13 of the U. S. Army Corps of Engineers, the Department of Commerce Motor Vehicle Conservation Task Group, the American Society of Refrigerating Engineers, the American Society of Naval Engineers, the American Ordnance Association, and the Society of American Military Engineers. He also is a member of the Lockport, New York, Chamber of Commerce and Community Fund Council.

Mr. Holmes is a registered professional engineer in the States of Michigan, New York, and Ohio.



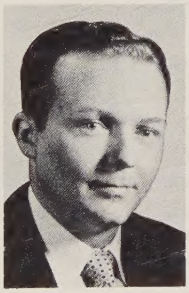
ROBERT W. LEWIS,

who prepared the problem "Determine the Maximum Tensile Stress in a Jet Aircraft Turbine Shaft with Over-hung Turbine Wheels" and the solution appearing in this issue, is a project engineer in the Engineering Department of the Aircraft Engines Operations at Allison Division, Indianapolis, Indiana. Mr. Lewis' duties in this position are concerned principally with stress analysis problems in the design of axial flow power sections for turbo-prop engines. He has been performing this work since joining Allison in July 1954 after completion of requirements for the M.S. degree at Iowa State College, Ames, Iowa.

While engaged in his graduate studies at Iowa State College, Mr. Lewis served as a research assistant for the Department of Aeronautical Engineering and the Institute for Atomic Research. His thesis subject was the tensile properties of metals in the inelastic range of stress. His previous industrial experience included two years at McDonnell Aircraft Corporation, St. Louis, Missouri, where his work was devoted to the design and stress analysis of high speed aircraft wing structure with emphasis on thick skin analysis. Mr. Lewis was awarded the B.S.M.E. degree in 1950 by the University of Missouri, Columbia, Missouri.

His technical society affiliations include membership in the American Society of Mechanical Engineers, Institute of the Aeronautical Sciences, and Society for Experimental Stress Analysis.

Mr. Lewis served with the U. S. Navy from 1945 to 1946.



ROBERT A. MOREAU,

co-contributor of "Field Testing of Diesel Locomotive Axles," is a project engineer in the Engineering Department, Electro-Motive Division, LaGrange, Illinois.

He first came to Electro-Motive after receiving the B.S. degree in applied mathematics and mechanics from the University of Wisconsin in June 1950. After working through the summer he returned to the University of Wisconsin, which had awarded him a fellowship, and obtained his M.S. degree in the same field of study in June 1951.

Returning to Electro-Motive, he has been involved in theoretical and analog computations of shock and vibration studies and in obtaining experimental stress information from various field tests, such as described in this paper. Currently under study is the cause of railroad wheel stress.

He is a member of the honorary societies Phi Beta Kappa, Phi Kappa Phi, Pi Mu Epsilon, and Phi Eta Sigma. In addition, he is a member of the Society for Experimental Stress Analysis. Some of the material in this paper was also the subject of a paper, co-authored with Ludvig Petersen, entitled "Field Testing of Railroad Axles," which he presented before the S.E.S.A. in 1954.

Mr. Moreau served in the United States Naval Reserve from July 1944 to June 1946, during which time he attended Wright Junior College, Chicago, Oklahoma A. and M., and Navy Pier (Electronics School), Chicago, graduating as an electronic technician. The remainder of his service was in the Asiatic Theatre.

LUDVIG PETERSEN,

co-contributor of "Field Testing of Diesel Locomotive Axles," serves as chief structural engineer in the Engineering Department of Electro-Motive Division, LaGrange, Illinois.

Mr. Petersen's career in the field of structural design spans nearly three decades in American industry. A native of Denmark, he was a naval architect in that country during the early 1920's. In 1923 he came to the United States, where his first technical work was as a detailer for Westinghouse Electric Company in East Pittsburgh and later as a customer's inspector at American Bridge Company in Gary. By 1926 he was in charge of a group of bridge designers in a Chicago organization and supervised the design of several bridges in Chicago which are still in use. He later worked for the Illinois Central Railroad, The Pullman Standard Car

Members of the engineering faculty of colleges and universities may receive the GENERAL MOTORS ENGINEERING JOURNAL regularly upon their request. Classroom quantities are also supplied regularly on request.

Manufacturing Company, and several Chicago contractors—always as a structural designer on bridges, buildings, and railroad equipment.

He came to Electro-Motive in 1936 where he was soon put in charge of a group of calculators. During the 18 years Mr. Petersen has been in charge of this group, it has grown along with the Division but its functions have changed with the general trend. From doing almost purely theoretical work, the weight has been shifted more and more toward replacing the time-consuming and not always reliable theoretical work with experimental work. It was in line with this thinking that the test was undertaken which is reported in his current paper.

Mr. Petersen's career is reflected in his affiliation with technical societies. He became an associate member of the American Society of Civil Engineers in 1927. Later, he became a member of the American Society of Mechanical Engineers and he has now for some years been a member of the Society for Experimental Stress Analysis.

Material presented in this paper was the subject of a presentation made before the S.E.S.A. in 1954.

WILLIAM S. PETTIGREW,

contributor of "The Importance of Liaison Between Engineering and Patent Departments" and co-ordinator of this issue's "Notes About Inventions and Inventors," has served as a patent attorney in the General Motors Patent Section since September 1950. He is assigned to the Patent Section's Central Office located in Detroit, Michigan. The other two offices are at Dayton, Ohio, and at Washington, D. C.

In his capacity as patent attorney, Mr. Pettigrew is charged with the preparation of patent applications for inventions developed by GM's employes, with the responsibility of making patent infringement investigations on new or changed devices which are used in production, and with the duty of determining the scope and validity of patents which appear to be pertinent to devices which are used in production. He is primarily concerned with those patent matters affecting the Fisher Body and Ternstedt Divisions and the General Motors Styling Section.

Mr. Pettigrew earned the A.B. degree from the University of Chicago in 1936. Two years later he received the J.D. (Doctor of Jurisprudence) degree from the University's Law School.

For the five-year period directly before joining General Motors, Mr. Pettigrew practiced patent law with the firm of Schroeder, Merriam, Hofgren, and Brady in Chicago. Prior to this, he served as a radio instructor for the Army Air Force Radio School, also located in Chicago.

A registered patent attorney, Mr. Pettigrew's professional affiliations include membership in the State Bars of Michigan and Illinois, and in the Michigan Patent Law Association.

WILLIAM E. SEHN,

contributor of "Testing the Structure of an Automobile Body," is senior project engineer on weatherstrips and rubber parts at the Fisher Body Division, Detroit, Michigan.

Mr. Sehn's entire experience in body engineering work has been with Fisher Body Division, starting in 1939 when he was employed as a student engineer and enrolled in General Motors Institute. He was graduated from the Institute in 1943 and was awarded the Bachelor of Mechanical Engineering degree in 1951. In 1944 he entered the U. S. Navy where his entire experience was concerned with engineering. He served as ship superintendent at the Norfolk Naval Shipyard and later in the Pacific and Far Eastern Theater, separating with the rank of lieutenant.

In 1947 Mr. Sehn returned to Fisher Body entering the Experimental and Development Section where, soon afterward, all of his work centered about the Structures Department and the physical testing of automobile bodies and their components. He continued in this capacity until 1951 when he was made senior project engineer in charge of glass and standard parts. He was transferred to his present post in 1954.

He is a member of several technical committees in General Motors and is a member representative of Subcommittee No. 3 of the American Standards Association Sectional Committee B-18, in addition to numerous subgroups on standardization of screws.

A registered professional mechanical engineer, Mr. Sehn is a member of the National Society and Michigan Society of Professional Engineers. He also is a member of the Engineering Society of Detroit where he has actively served on several committees.

Contributors' backgrounds vary greatly in detail but each has achieved a technical responsibility in the field in which he writes.

JOHN M. WETZLER,

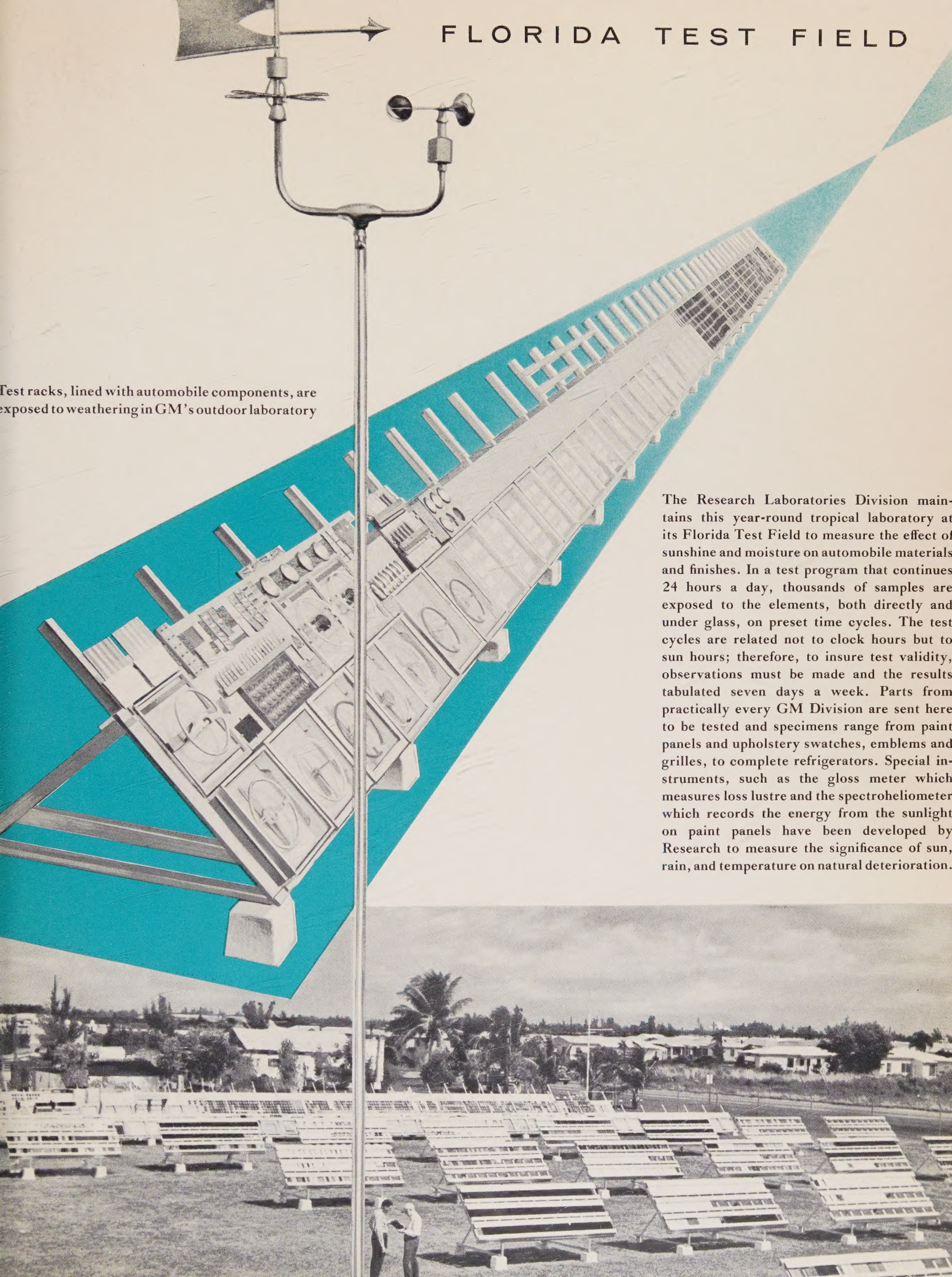
co-contributor of "Analyzing a Typical, Simple-Cycle Turbo-Jet Design and Development Program," serves Allison Division, Indianapolis, Indiana, as a section head in its Turbo-Jet Design Group.

Mr. Wetzler's entire career since receiving the B.S degree in mechanical engineering from The Johns Hopkins University in 1939 has been with General Motors Corporation. He started with the GM Proving Ground, Milford, Michigan, shortly after graduation, as a junior engineer in its Engineering Test Department. This experience in development testing was broadened by his transfer to Allison's Testing Department in 1941 where he remained for the next four years.

In 1945 he was promoted to supervisor of the Parts Test Laboratory at that Division and in 1946 he became supervisor of piston-engine test projects. Since 1947 Mr. Wetzler has served as head of the Turbo-Jet Aero-Thermodynamics Group and later head of the Aircraft Installation Group before promotion to his present position in 1954. As a section head in the Turbo-Jet Design Group, his current work involves the development of advanced J71 turbo-jet engines.

Mr. Wetzler was elected to the honorary society Tau Beta Pi and for the past three years has served on the National Advisory Committee for Aeronautics' Subcommittee on Compressors and Turbines.

FLORIDA TEST FIELD



Test racks, lined with automobile components, are exposed to weathering in GM's outdoor laboratory

The Research Laboratories Division maintains this year-round tropical laboratory at its Florida Test Field to measure the effect of sunshine and moisture on automobile materials and finishes. In a test program that continues 24 hours a day, thousands of samples are exposed to the elements, both directly and under glass, on preset time cycles. The test cycles are related not to clock hours but to sun hours; therefore, to insure test validity, observations must be made and the results tabulated seven days a week. Parts from practically every GM Division are sent here to be tested and specimens range from paint panels and upholstery swatches, emblems and grilles, to complete refrigerators. Special instruments, such as the gloss meter which measures loss lustre and the spectroheliometer which records the energy from the sunlight on paint panels have been developed by Research to measure the significance of sun, rain, and temperature on natural deterioration.

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